POWER

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Keep Out of the Rut

SO MUCH has been said and written about keeping out of ruts that any further discourse upon the subject must seem bromidic and trite.

And yet I am seeing so much of it, such persistence in doing things in the way that they have always been done, such inertia, reluctance, prejudice, bigotry, against change of method or view, that I can well understand the necessity for the eons of time required for the gentle processes of evolution.

There seems to be, with respect to many things, a feeling akin to that which makes a man put on his left shoe first because he has been prospering pretty well doing it in that way, and if he should change he might derange the program and upset the equanimity of the propitious influences to which he ascribes his success.

And then it takes an effort to get out of a rut, which may give one considerable of a jolt. It is a lot easier to keep on in the other fellow's tracks. You will get to the same place that he did—somewhat later.

But the fellow who pulls up out of the rut and goes along, especially the fellow who looks farther and wider than a few rods of road ahead and gets an intelligent idea of the lay of the land and traveling conditions, can often drive all around the plodder in the rut and meet him on the way home.

I have seen many a man started upon a successful career by being kicked out of the rut into which he had settled by something that, at the time, he considered an irreparable misfortune.

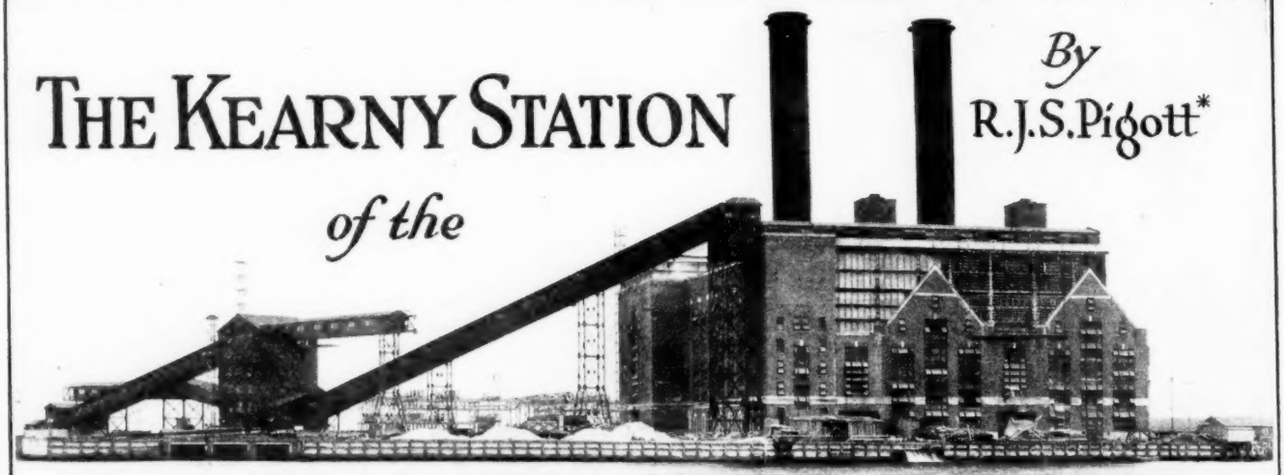
But the idea that I had in mind when I started this was that, aside from its effect upon the driver, running in ruts is bad for the ruts themselves.

Every fellow who hangs to the rut makes it deeper and muckier and harder to get out of.

Take account of the national habit of thought with respect to questions of politics, religion, race, business ethics, social economy, the co-ordination and development on sound scientific, business and economic principles of the natural resources and the manpower of the nation, and see if progress is obstructed by physical difficulties or by the ruts of prejudice, superstition, inertia and smug satisfaction with things as they are.

THE KEARNY STATION of the PUBLIC SERVICE ELECTRIC POWER COMPANY

By R. J. S. Pigott*



SOUND and conservative engineering is evident in the design of the latest plant of the Public Service system. The present installation of 185,000 kw. is half of the ultimate plant. Operating on the regenerative cycle, the plant relies for efficiency on superior operation of simple equipment rather than on the addition of special heat recovery devices.

THE Kearny Power Station is situated on the west bank of the Hackensack River in Kearny, New Jersey, on the meadow land between Jersey City and Newark, at a point approximately equidistant from Essex and Marion, two other stations operated by the Public Service Electric & Gas Company. The site, which is near the load center of the district, was selected because an ample supply of condensing water is available and coal can be brought in by barge or by rail over the main line of the Pennsylvania Railroad. The plant is designed to burn high-grade bituminous coal of about 13,500 B.t.u. per lb. as fired.

The coal bunker is constructed on the river bank, paralleled by the boiler house, turbine room, switch-house, administration building and outdoor substation in the sequence mentioned. This order was decided upon because it was felt that most of the coal would be delivered by barge and because there was no particular difficulty in constructing the more extensive circulating water canals required.

All the foundations are built on a heavy concrete mat resting on wooden piles. The circulating water canals, located below this mat, are of reinforced concrete and their vertical walls act as ribs for the mat, distributing the load properly.

All the turbines are supplied with steam at a pressure of 340 lb. per sq.in. abs. with a total temperature of 700 deg. F. Three turbines are rated at 35,000 kw. and two at 40,000 kw. The two 40,000-kw. turbines are equipped with Baumann blading in the exhaust end. All the turbines have water seals on both the high-pressure and the low-pressure shaft packings.

The unit system of turbine oil purification has been incorporated. Each turbine has its own oil reservoir, clean oil tank, impure oil tank, centrifugal oil purifier, and return oil pump. However, particular attention has been given to flexibility, and in an emergency any turbine may operate on the oil system of any other turbine.

The turbine oil coolers, which operate normally with fresh service water, have sufficient surface to give satisfactory results with cooling water entering at temperatures as high as 90 deg. F.

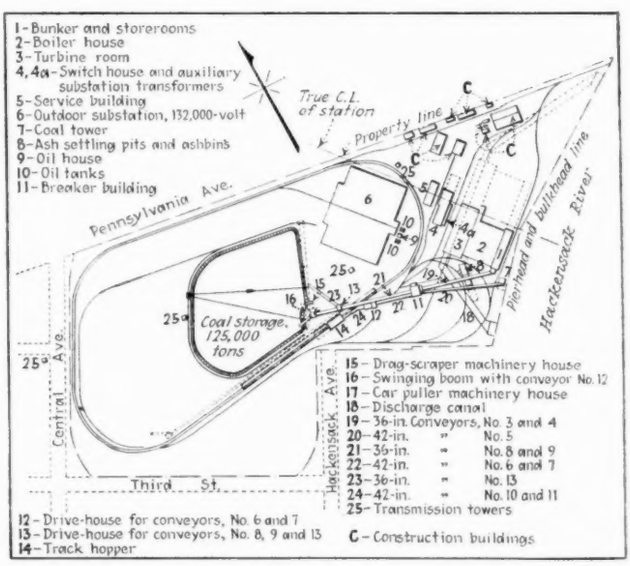


Fig. 1—The dotted lines show the future plant extension, with a double-ended discharge canal

*Consulting Mechanical Engineer of the Public Service Production Company, designers and builders of the plant, which is owned by the Public Service Electric Power Company and operated under lease by the Public Service Electric & Gas Company.

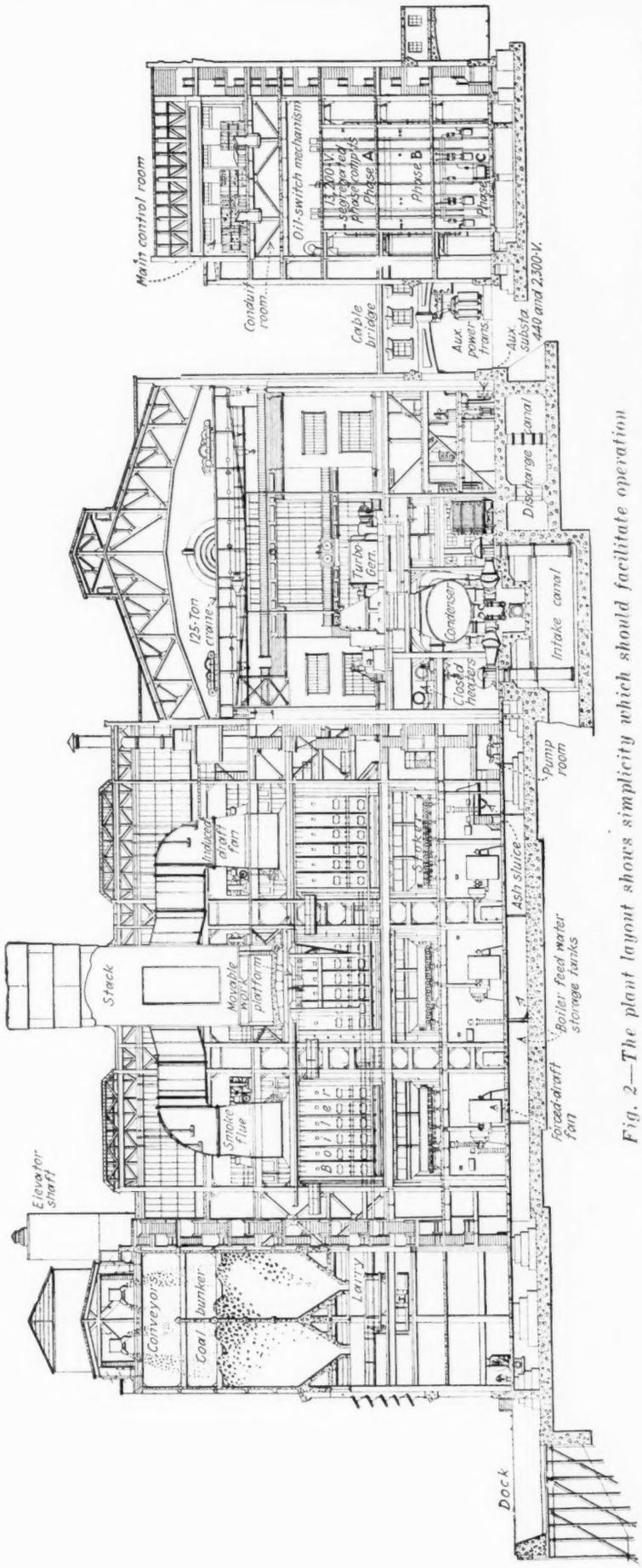


Fig. 2—The plant layout shows simplicity which should facilitate operation

The three-phase generators operate normally at 13,800 volts, and each generator has twelve thermocouples embedded in its windings to indicate the temperature in various portions. Cooling is done by air in a closed system. No attempt is made to recover any of the generator losses because it is felt that in a plant employing stage bleeding for feed-water heating the small heat recovery hardly compensates for the extra cost and complication. Salt water for cooling the ventilating air is taken directly from the discharge of the circulating pumps. Sufficient surface has been provided to remove the generator heat losses at maximum load with the cooling water entering at 83 deg. F. The water supply is cross-connected so that water can be taken either from the circulating pumps on the unit to which the cooler is connected, or from a common header connecting the circulating pumps of all units. This latter connection is to be used whenever a generator is disconnected from its turbine and operated as a synchronous condenser.

In case of a failure of the circulating water supply, weighted air dampers open automatically and convert the closed system into an open system. The dampers are normally held closed by air cylinders which operate when the pressure of the water passing through the cooler drops.

Several New Features in Condensing Plant

Each turbine is served by a 50,000-sq.ft. surface condenser, with divided water boxes so that one side can be cleaned while the other side is operating. The condensers are bolted directly to the turbines and are supported on springs. Rubber expansion joints are provided in the circulating water and condensate connections.

Each condenser is provided with two motor-driven circulating pumps so connected that any pump can supply either half of its own condenser or either half of any other condenser in the station. There is also installed at the temporary end of the present building, which will ultimately be the center of the completed station, one large turbine-driven circulating pump for use with any condenser.

Two batteries of steam-jet air pumps are provided for each condenser. Each battery consists of two pumps mounted on a common intermediate and after condenser. Three of the condensers have hydraulic air pumps as additional equipment.

Duplicate condensate pumps are provided for each unit—one motor driven, the other steam driven.

Intake and Discharge Canals

The intake canals extend from the face of the dock, past the bunker and boiler-house foundations to the suction chambers beneath the turbine foundations. One canal is provided for each condenser, and a sixth canal

has been installed as part of the original installation for construction reasons. They are of rectangular cross-section, 9 ft. 6 in. high and 6 ft. 6 in. wide. This gives a water velocity of 3.2 ft. per sec. when furnishing 88,000 gal. per minute.

The Hackensack River is subject to flow and tide variations, and to secure a continuous supply of water irrespective of the tide level, the underside of each intake canal roof has been put two feet below the lowest recorded level. To prevent obstruction of water flow occasioned by coal barges moored at the dock, the face of the canals is brought back four feet from the dock line.

Each intake canal is provided with a trash rack at the entrance and with revolving screens in the bunker-house basement. Each screen well can be isolated by motor-operated sluice gates for cleaning and repairs.

The five generating units discharge their circulating water into chambers beneath the condensers. Each of these spills into a main discharge canal over a weir, which insures that the condenser tail pipes will always be submerged irrespective of water levels elsewhere in the system. The discharge chamber floors and the weirs are protected against erosion by heavy cast-iron plates.

The discharge canal is proportioned to carry off the discharge and service water from the ultimate station at a velocity of 3.8 to 7.5 ft. per sec. depending upon tide conditions. The cross-section of this canal is graduated to accommodate the increasing water quantity and terminates at the dock in a forebay proportioned to reduce the exit velocity.

The ultimate arrangement is to discharge the water into the river either upstream or downstream as may be required to eliminate recirculation. At present, only the downstream section of the canal system is built, but when the station is extended, the other discharge channel will be built and discharge control will be secured by the use of balanced segmental gates.

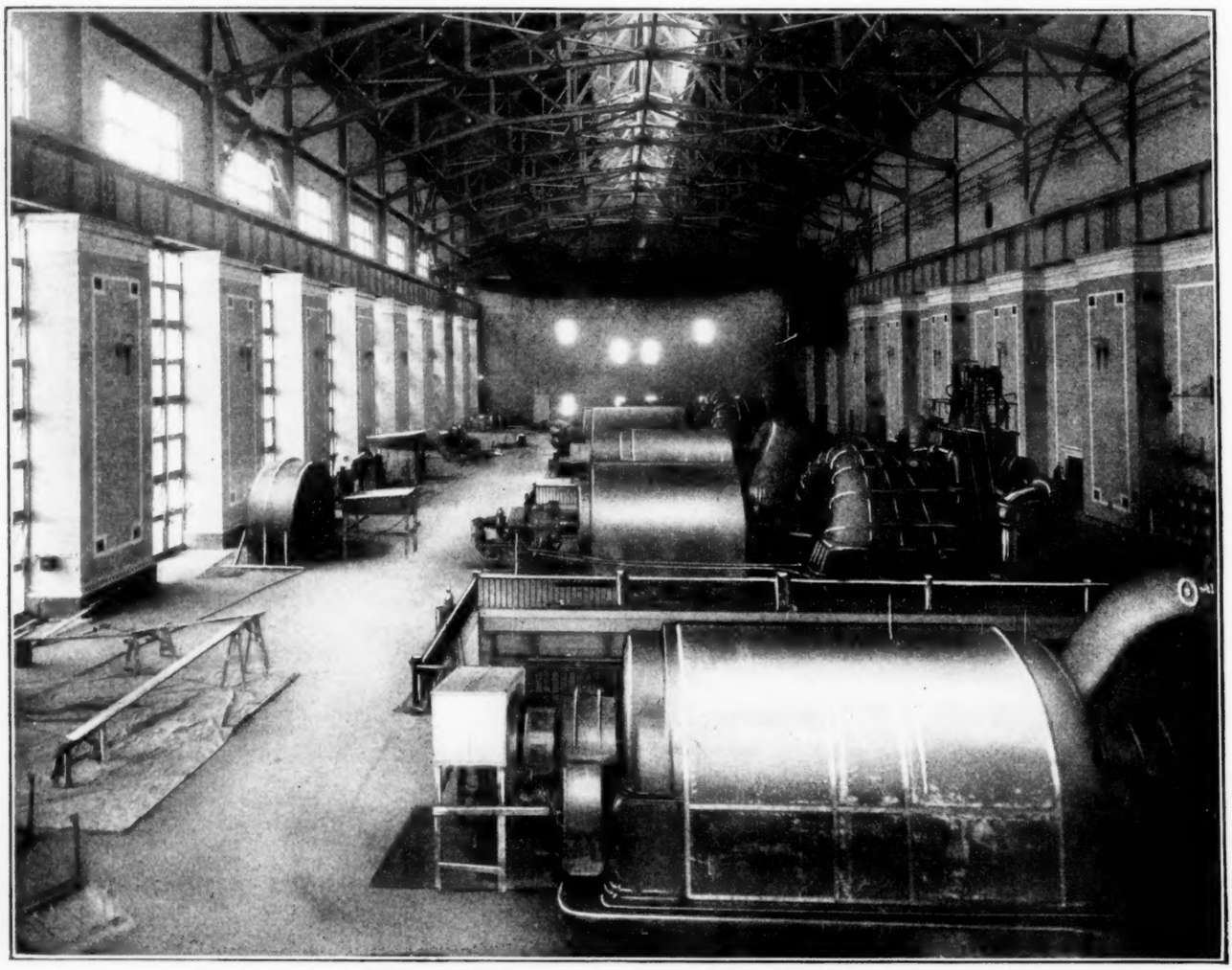


Fig. 3—Large hatches admit light to the condenser room, and simple architectural elements make the turbine room attractive

Stoker-Fired Boilers Without Economizers or Air Preheaters

The present installation consists of twelve cross-drum boilers. The heating surface of each boiler is 23,600 sq.ft., and each superheater has an area of 4,130 sq.ft. The boilers have 48 sections each, 20 tubes high and 21 ft. long. The superheater is between the upper and lower decks of the boiler with six tubes between the superheater and the furnace.

The boilers are cross-connected in such a manner

that no particular set of boilers serve a particular turbine, the boiler room being arranged in four sections, while the turbine room has five units in the present installation.

In order to secure high efficiencies when operating above 200 per cent of rating (6,700 B.t.u. per sq.ft. per hour), a large furnace volume with a setting height of 22 ft. was adopted. Owing to the height of the

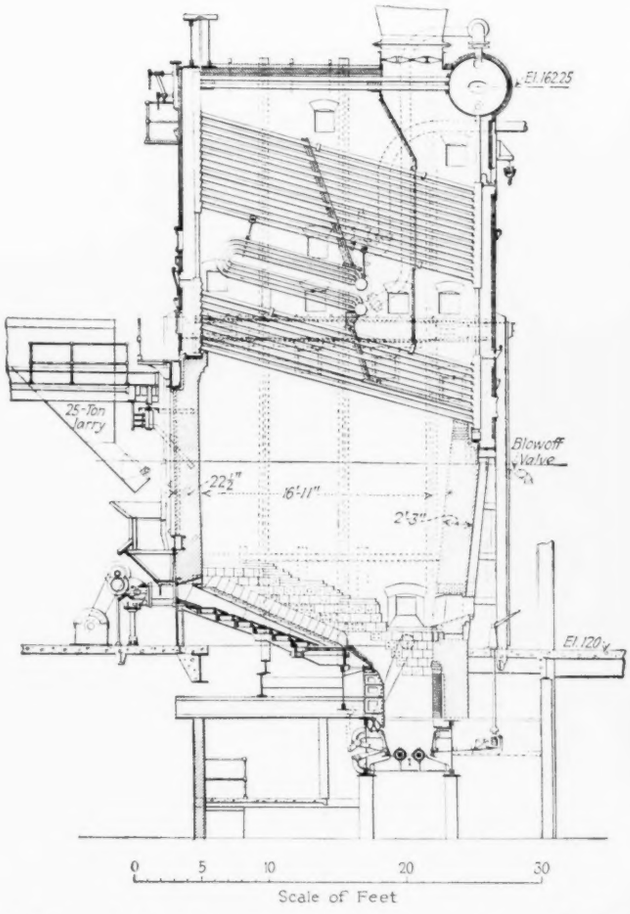


Fig. 4—A high setting, with stokers of record size, should give large capacity and high economy

setting the upper part of the structure opposite the boiler tubes is supported on structural steel separately from the lower or furnace part.

The walls of the setting are air cooled. The furnace side walls and the rear wall are lined for a short distance above the grates and clinker pit with non-clinkering carborundum blocks. Those in the lower rows are perforated, and cooling air is supplied to them from the forced-draft air duct. The front wall is also lined just above the stoker with carborundum brick. These are solid, but are backed up by a pocket receiving air from the forced-draft duct and discharging to the fanroom.

Above the traveling weigh larries an adjustable traveling platform is provided in each firing aisle. This will be used in cleaning boiler tubes, and work can be carried on without interfering with the larries.

The stokers are of the single-fired type with clinker grinders, driven by four-speed constant horsepower induction motors together with two-speed power boxes, so that ratings up to 350 per cent can be secured.

The coal-handling system uses inclined belt conveyors for conveying coal from the coal tower and the car-unloading pit to either the yard storage or the coal bunkers in the station.

Coal is hoisted from barges by means of a motor-driven grab-bucket hoist having a capacity of 350 tons an hour. After passing through the breaker, the coal is delivered either to the yard storage or to the coal bunkers. The coal bunkers hold 7,500 tons, which is a sufficient supply for more than six days. Two motor-driven traveling weigh larries, each having a capacity of 25 tons and a speed of 150 ft. per min., take the coal from the bunkers and deliver it to the extension hoppers on the stokers, which hold 9 tons each.

In the yard storage a single drag scraper is installed at present, which gives a yard capacity of 150,000 tons in 15-ft. piles. When the station is extended, two or

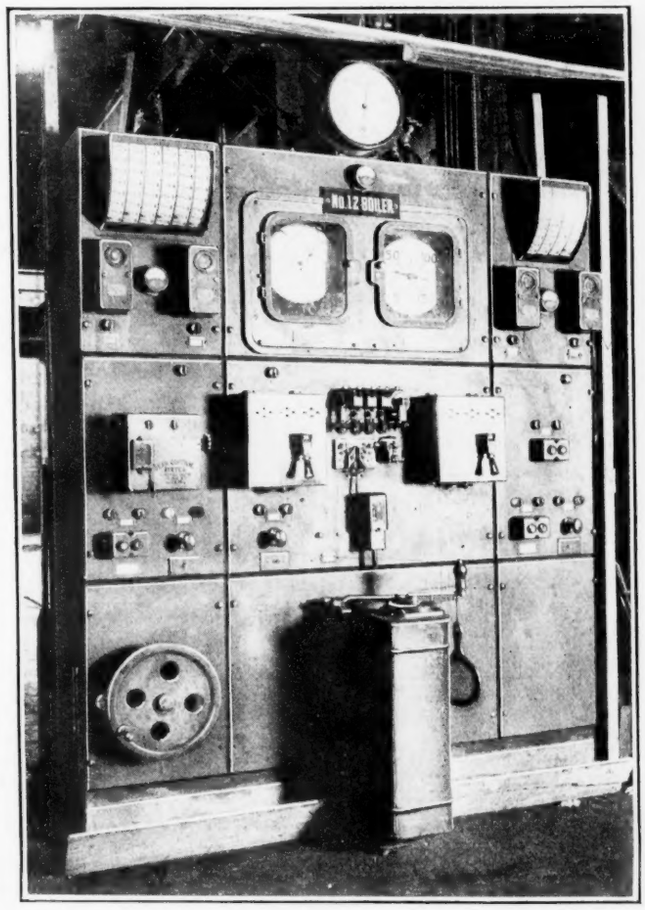


Fig. 5—Boiler instruments and controls help the operator secure high efficiency

more additional drag scrapers will be installed, which will increase the storage to 275,000 tons. An additional 90,000 tons can be cared for by locomotive cranes.

In the completed station all the belt conveyors, breakers and car unloading pits will be in pairs, with the exception of the belt from the coal tower to the breakers. The yard belts are made reversible and the hoppers and chutes are so arranged that either belt can be used for storing or reclaiming.

The ashes are dropped into a sluiceway and washed to a settling pit at the end of the building. The sluiceway below the water line consists of cast-iron sections

protected by hard, white cast-iron liners having a high resistance to the abrasive action of the ash in the water. Above the water line, an interlocking tile lining is provided. All parts can be replaced in case of excessive wear.

Two sluicing pumps are provided supplying 733 gal. per min. of sluicing water per boiler under a head of 115 ft. A spare pump is also provided.

The ash settling pit outside the building has a capacity of 600 cu.yd., which is sufficient for more than one day's ashes from the finished station. A gantry crane with a grab bucket is installed over the settling pit, and ashes can be loaded either into railroad cars or into trucks. Storage hoppers are provided over both the railroad track and the roadway, so that the crane can be operated continuously without waiting for the placing of cars or trucks.

With the exception of a few stand-by units all auxiliaries are motor driven. No house turbines are installed, the power for the auxiliaries being taken

Most of the motors are remote controlled and of the double-squirrel-cage type. Multi-speed operation is used for the fan and stoker drives. The motors for the forced-draft and induced-draft fans are thrown directly on the line by remote-controlled breakers. Pole-changing switches, providing for the operation of each fan motor at two speeds, are located at the motors and

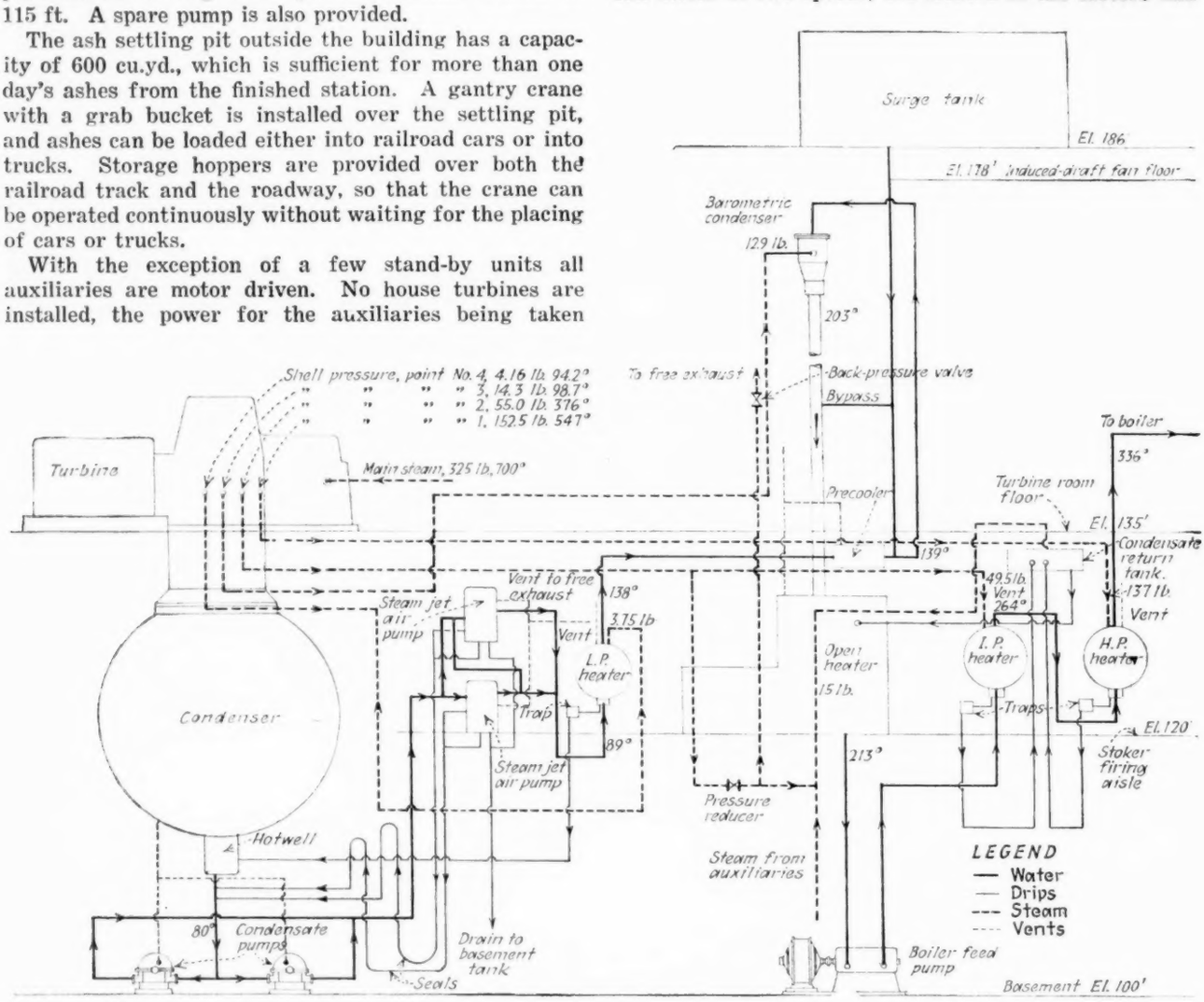


Fig. 6—Four-stage bleeding reduces heat losses, and an atmospheric heater deaërates the water and makes possible a boiler-feed-pump suction header

from the main buses through step-down transformers. The plant is divided electrically into two sections. The first consists of two 39,200-kva. and one 43,750-kva. generators; the other group of two generators, one of each of these ratings. Both these groups are tied to the main bus in such a way as to insure service to the station auxiliary transformers whether one or both of the electrical sections is furnishing power. The main tie buses are in duplicate.

The station power, with a few exceptions, is three-phase. Motors of 50 hp. and over are 2,300-volt, and smaller motors are 440-volt. Three 7,500-kva. transformer banks supply the 2,300-volt buses. The power for the 440-volt buses is stepped down from 2,300 volts by six banks of transformers. Two of these banks supply the smaller motors in the turbine room and boiler house; four supply the power for coal handling.

are controlled by push buttons. Intermediate adjustment of fan output is obtained by means of dampers. The stoker motors have double stator windings and these, together with remote-controlled pole-changing switches, permit operation of each motor at four speeds, as already mentioned. The two-speed power boxes give eight stoker speeds.

The circulating pump motors are of the slip-ring type with special starting and under voltage features. When any line switch is closed, the corresponding motor is automatically brought up to speed by a motor-operated drum switch which controls contactors short-circuiting the secondary resistance. Each motor is so connected as to stay on the line under all low-voltage conditions, two steps of secondary resistance being cut back in at 50 per cent voltage and the entire secondary resistance at 10 per cent voltage. The line

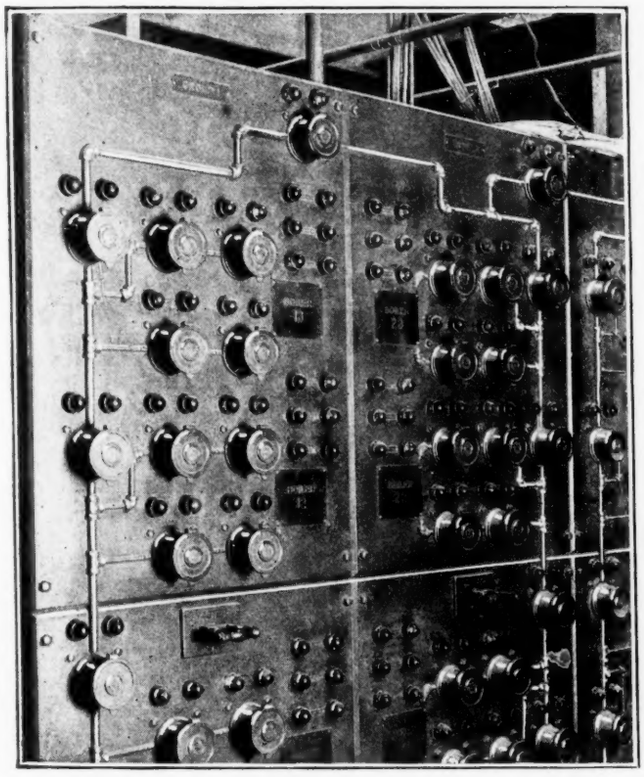


Fig. 7—A glimpse of the valve control board

breaker is not opened unless an overload relay closes or the operator in charge trips it.

Synchronous motors with squirrel-cage starting windings drive the air compressors. Each motor is thrown directly on the line, and a relay, acting after a given period of time, causes the motor field to be energized.

The motors driving the boiler-feed pumps are of the slip-ring type. The speed of the motors is controlled by differential-pressure regulators applied to these controllers.

The feed water returning to the boilers is heated with steam bled from four stages of the main unit. As the stage pressures in the turbine vary both with the load and with the make of the turbine, and as it is considered necessary to raise the temperature of the feed water to 212 deg. F. in order to reduce the oxygen in the water in open heaters by counterflow of steam, barometric heaters are provided at the third bleeding connection. These consist of barometric condensers discharging hot water into open heaters. Under average load conditions the temperature of the water in the open heaters would be below 200 deg. F. if dependence were placed solely upon the steam taken from the bleeding connection. However, there are auxiliary connections with reducing valves, from bleeding points at higher pressures from which additional steam can be secured to boost the temperature in the open heaters to the requisite 212 deg. F.

This arrangement permits the use of a common feed-pump suction header, and also permits the pressure on the bleeding connections to vary in any manner without affecting the operation of the feed pumps.

Precautions are taken to avoid the reverse flow of steam and water in the bleeder piping. Starting at the turbine, each of the four bleeder lines contains a hand-operated gate valve, an emergency solenoid-operated gate valve, and a check valve. The three closed heater lines are also fitted with spill valves.

In case the main turbine throttle valve trips out, the solenoid-operated gate valves in all the bleeder lines are automatically closed and can be reset only after the turbine throttle valve has been opened.

Each closed heater is equipped with a float. If a tube should break, or if for any other reason the water level in the heater rises beyond a safe point, the float will energize the solenoids on the spill valve and the emergency gate valve in its particular bleeder line, opening the former and closing the latter. A water relief valve is also provided in case the spill valve fails to operate.

Centralized Emergency Valve Control Is a Feature of the Piping Installation

The centralized emergency valve control board is placed on the bridge between the turbine room and the switchhouse at the center of the ultimate station. This location was selected not only as a point centrally situated, but also as one with little possibility of being enveloped in steam in the event of a break in the steam lines. The board has mounted on it a mimic system showing the main steam piping, boilers, turbines, and valves; and is equipped with emergency closing stations, indicating lamps, and emergency group-closing drum controllers.

The closing stations permit the emergency closure of individual valves. Each master controller permits the emergency closure of all steam valves of a given group in the event of the failure of the individual

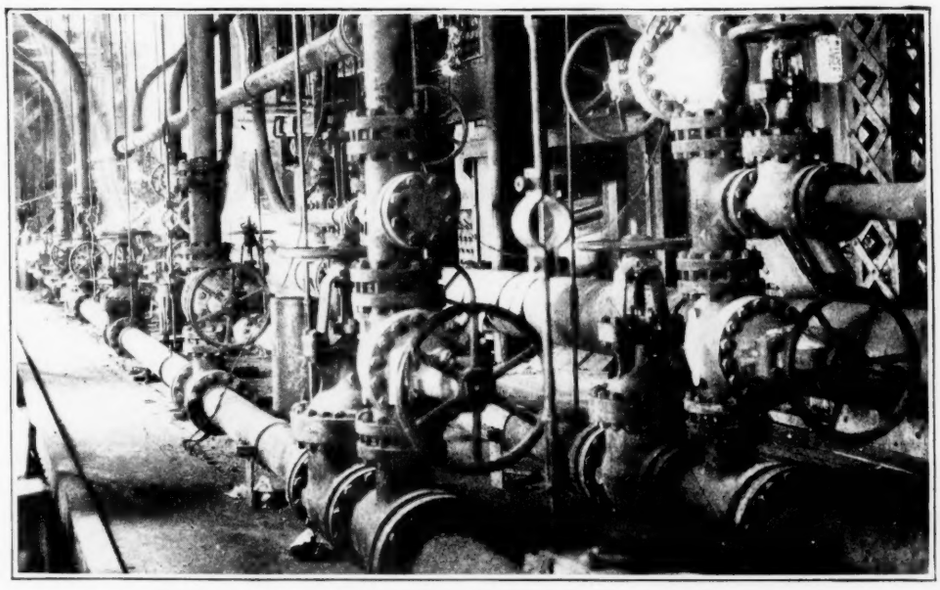


Fig. 8—Piping behind the boilers is arranged to make valves easily accessible from suitable platforms

closing stations to isolate the break. The master controller also trips out the forced-draft and induced-draft fans and the stoker drives, but does not trip out the clinker-grinder drives.

In addition to the centralized emergency control of the main steam valves, local opening and closing stations have been provided near the individual valves.

In the main and auxiliary steam lines as well as in the boiler feed piping, Sargol welded joints are used for all lines 4 in. or more in diameter. From 2 to 3½ in. the pipe lines are made up with tongue-and-groove joints and copper gaskets. The high-pressure piping is lap welded. All nozzles are of the hammer-welded flue type except that for the 20-in. pipe they are of the saddle type.

The insulating covering for the high-pressure steam lines consists of a layer of high-temperature material next to the pipe surface of sufficient thickness to reduce the temperature to below 600 deg. F. The outer covering is standard 85 per cent magnesia.

MECHANICAL EQUIPMENT—INITIAL INSTALLATION, KEARNY STATION

Total Capacity 205,100 Kva.

TURBINE-GENERATORS

	Nos. 1, 3 and 5	Nos. 2 and 4
Manufacturer	General Electric Co.	Westinghouse Elec. & Mfg. Co.
Steam pressure at throttle, lb., gage	325	325
Superheat at throttle, deg.	271	271
Stage bleeding points	4	4
Rated capacity, kva. at 80% power factor	39,200	43,750
Generator	13,800-v. star-connected 3 ph., 60 cy.	13,800-v. star-connected 3 ph., 60 cy.
Ventilation of generator	Griscom-Russell U-Fin Coolers. Total surface, 18,240 sq. ft. per generator	Griscom-Russell U-Fin coolers. Total surface, 22,800 sq. ft. per generator
Cooling air c.f.m.	90,000	105,000
Cooling water (salt) g.p.m.	1,450	1,500
Maximum permissible temp. water to cooler deg. F.	84	83
Direct-connected exciter	150 kw., 250 v.	165 kw., 250 v.
Lubrication	One geared pump and one turbine-driven auxiliary pump with automatic pressure control	One geared pump and one turbine-driven auxiliary pump with automatic pressure control
Oil cooler	Andale	Griscom-Russell
Maximum permissible inlet temp. water to oil cooler, deg. F.	90	90
Overall length including exciter	51 ft. 6⅝ in.	61 ft. 5½ in.
Overall width	19 ft. 6½ in.	22 ft. 0 in.

SURFACE CONDENSERS ON UNITS

Manufacturer	Nos. 1 and 3 Allis-Chalmers Mfg. Co.	Nos. 2 and 4 Westinghouse Elec. & Mfg. Co.	No. 5 Wheeler Cond. & Eng. Co.
Total tube surface, sq. ft.	50,000	50,000	50,000
Number of passes	2	2	2
Supports	Springs	Springs	Springs
Connection to turbine	Rigid	Rigid	Rigid

CIRCULATING PUMPS (2 per condenser)

Capacity each, g.p.m.	38,000	38,665	38,000
Discharge head (2 pumps operating), feet	21	21	21
Maximum suction head, ft.	20	20	20
Speed, r.p.m.	353	345	345
Motor drives, hp.	500	500	500

CONDENSATE PUMPS (2 per condenser)

Capacity, each, g.p.m.	1,000	1,000	1,000
Total head including vacuum, ft.	175	175	175
Motor drives, hp.	100	100	75
Turbine drives, hp.	100	85	75
	Terry	Westg.	Westg.

STEAM-JET AIR PUMPS (2 per condenser)

Make	Croll-Reynolds	Westg.-LeBlanc	Wheeler
Inter-condensers per surface condenser	2	2	2
Rated capacity free air per twin unit to 28½-in. vac. (2 units per surface condenser), c.f.m.	40	41.2	44
Condensing medium in inter and after coolers	Condensate	Condensate	Condensate

HYDRAULIC AIR PUMPS

	Westinghouse	Wheeler Cond. & Eng. Co.
Number per condenser	1	1
Speed, r.p.m.	700	1,775
Motor drives, hp.	75	75
Capacity free air at 0.76 in. Hg. partial air press., 70° hurling water c.f.m.	20	30
Hurling water g.p.m.	1,330	500

SPARE STEAM-DRIVEN CIRCULATING PUMP

One 38,665 g.p.m. Westinghouse
Discharge head ..... 21 ft.
Maximum suction lift ..... 20 ft.
Speed ..... 345 r.p.m.
Drive, 500-hp. Westinghouse M.D. steam turbine at 2,850 r.p.m. through an 8.27 to 1 reduction gear

TURBINE LUBRICATING OIL PUMPS AND PURIFIERS

7—DeLaval back-geared pumps, with 5 No. 600 DeLaval centrifugal oil purifiers, driven by G.E. motors

BOILERS AND AUXILIARIES

Boilers	12 Babcock & Wilcox Co.
Type	Cross drum
Steam-making surface per boiler, sq. ft.	23,600
Arrangement of tubes	20 tubes high by 48 wide. Lowest two tubes lowered to form a slag screen
Boiler setting height	22 ft. with firing from the nigh end
Maximum steam pressure	385 lb. gage
Make of superheater	Babcock & Wilcox
Superheater surface per boiler	4,130 sq. ft.
Location of superheater	First pass above sixth row of tubes
Total steam temperature at 200 per cent rating	720 deg. F.
Ratio superheater surface to steaming surface	0.175
Soot blowers	Diamond Power Specialty Corp. 22 heads per boiler
Blowoff valves	Crosby Fulway valves in tandem with Lunkenheimer angle-type valves
Feed regulators	Two 4-in. per boiler. Northern Equipment Co. (Copes), tension type

STOKERS

Manufacturer	{ Sanford-Riley Stoker Co. / American Engineering Co. (Taylor)
Drive, each stoker	One 25-hp. four-speed G.E. motor with Morse chain drive
Clinker grinder drive each stoker	Two 5-hp. single-speed G.E. squirrel-cage motors with Morse chains, 1,200 r.p.m.
Coal agitator	Negus Tiffany

	Riley	Taylor
Number of retorts	15	16
Ratio speeds—two-speed power boxes	1.75 to 1	1.5 to 1
Furnace width	28 ft. 4 in.	28 ft. 4 in.
Clinker grinder rolls per stoker	1	2
Furnace vol. including first pass, cu. ft.	8,985	8,472
Ratio water-heating surface to total projected area grate and clinker grinder	49.25	51.3
Ratio furnace vol. to water-heating surface	0.381	0.359
Ratio furnace vol. to projected grate and clinker grinder area	18.75	18.45

STACKS

Number	2 installed at present
Boilers per stack	6
Type	Riveted steel plate, brick lined to top, supported on building steel
Height above firing-room floor (el. 120 ft.)	258 ft.
Diameter inside	24 ft. 7 in.

MECHANICAL EQUIPMENT—INITIAL INSTALLATION, KEARNY STATION—CONCLUDED

FORCED-DRAFT EQUIPMENT

Manufacturer	Buffalo Forge Co.
Number fans installed	12,one per boiler,three per air duct
Capacity	134,000 c.f.m. at 65 deg. F. and 7 in. water pressure
Drives	225-hp., 2,300-v. two-speed G.E. motors

INDUCED-DRAFT EQUIPMENT

Manufacturer	B. F. Sturtevant Co.
Number installed	12 (motor driven) one per boiler, six per stack
Capacity	275,000 c.f.m., 650 deg. F. and 3½ in. water suction

AIR COMPRESSORS (2)

Manufacturer	Ingersoll-Rand Co.
Type	Horizontal cross-compound, two-stage with inter and after coolers
Capacity	520 c.f.m. free air
Air pressure	100 lb. gage
Drive	Direct connected 105-hp., syn. motor

BOILER-FEED PUMPS

Manufacturer	Allis-Chalmers Mfg. Co.
Number	3 motor driven and 2 steam-turbine driven
Number of pumps in normal serv.	One motor-driven pump for each two turbines, or three for present installation
Number of pumps normally spare	Two for present installation
Type	6-in., five-stage, centrifugal, single suction with hydraulic balance
Capacity	1,500 g.p.m. at 1,125 ft. total head 212 deg. F.
Control	Ruggles-Klingemann differential pressure regulators on the motor-driven pumps and Foster Engineering Co. differential pressure governor on steam-driven pumps

CLOSED FEED-WATER HEATERS

Type	Four-pass condenser type with floating head
Make	Griscom-Russell Co.

	High Pressure	Intermediate Pressure	Low Pressure Large	Low Pressure Small
Number of heaters	5	5	3	2
Tube surface, sq.ft.	2,000	2,000	1,700	1,400
Average steam press. abs., lb.	145	55	8	4
Av. temp. water entering, deg. F.	262	213	93	89
Av. temp. water leaving, deg. F.	322	262	163	133
Av. vel. water thru tubes, ft.-sec.	4.0	4.0	3.27	3.79

The drainage traps for the condensate on the closed feed-water heaters were special large-sized traps furnished by the Davis Engineering Co. These traps are all of the float type operating rotating ball valves

OPEN FEED-WATER HEATERS

Type	Cochrane Corp. deaerating and metering with jet heater
Number	5
Capacity	400,000 to 1,000,000 lb. per hr.

SERVICE PUMPS

Salt Water Service— Three 1,000 g.p.m. 250 ft. total head, Cameron pumps, direct connected to 125-hp., 1,800 r.p.m., 2,300-v. double-wound rotor type, G. E. motors. One of these pumps is equipped for fire service.

Fresh Water Service— Three 750-g.p.m., 175-ft. total head, DeLaval pumps, direct connected to 50-hp., 1,800 r.p.m., 440-v., double-wound-rotor type G.E. motors

PIPING

Main and auxiliary steam and boiler feed

Lap-welded pipe with cast-steel fittings designed for 400-lb. pressure, and 750 deg. F. temp. All nozzles are hammer welded flue type

Sargol welded Van Stone joints on sizes 4-in. and over

Tongue-and-groove joints with hammer-welded forged steel flanges on sizes 3½-in. to 2-in. inclusive

Sizes under 2-in. are fitted with forged steel unions with special groove, screwed on pipe and brazed

Pipe fabricated by the M. W. Kellogg Co.

Special steel bolts used throughout

Circulating Water— Cast-iron pipe and fittings furnished by Pittsburgh Valve, Foundry & Construction Co. and the Warren Foundry & Pipe Co.

Safety Valve Discharge— Lap-welded pipe with Van Stone joints and cast-iron flanges on sizes 8-in. and over. Standard screwed flanges on sizes under 8-in. Whitney MacDonald Co.

VALVES

Atmospheric relief	Atwood & Morrill 42-in. horizontal
Bleeder lines	Solenoid controlled trip and spill valves, Schutte & Koerting Co.
Boiler blowoff piping	Crane, H. P. cast-steel gate valves
Circulating water	42-in. angle check valves, Atwood & Morrill, 42-in. gate valves, Pittsburgh Valve, Foundry & Construction Co.
Main Steam	8-in. stop and check valves, Schutte & Koerting Co. H.P. cast-steel gate valves, Pittsburgh Valve, Foundry & Construction Co., and Schutte & Koerting Co.

VALVES—Concluded

Safety valves	Crosby Steam Gage & Valve Co.
Vacuum & condensate	Check and Globe valves, Crane Co
Closed heaters	Water relief valves, Crosby Steam Gage & Valve Co.
Motor control	Cutler-Hammer (Dean)
Boiler feed	Crane H.P. cast-steel gate valves

COAL-HANDLING SYSTEM

14, 36- and 42-inch belts, run on 5-pulley troughing idlers, furnished by Robins Conveying Belt Co.

Average speed of belts— 350 ft. to 400 ft. per minute. Belts furnished by Empire Tire & Rubber Co. and the Manhattan Rubber Co.

DRAG SCRAPER

Manufacturer	Maine Electric Co., capacity 400 tons per hr.
Length of haul	600 ft.

CAR HOPPER PIT

Four hoppers each with three discharges. Four cars will dump at one time. Twelve 36-in. belt feeders arranged in four batteries of three each

Horsepower required per battery of feeders 10

The present pit will handle 400 tons per hour; ultimate capacity, 800 tons per hour with twenty-four 36 in. feeders in operation

BREAKER HOUSE

Bradford breaker	Robins Conveying Belt Co.
Size	12 ft. diam., 22 ft. long
Capacity	400 tons per hr.
Drive	150 hp., 14 r.p.m.

COAL TOWER

Make	Mead-Morrison Mfg. Co.
Capacity of hoist	350 tons per hr.
Capacity of the bucket	5,500 lb.
Rope speed	620 ft. per min.
Total time for one trip	26 seconds

BUNKERS

Steel frame bunkers, lined with concrete

The bunker pocket opposite each firing aisle holds 3,800 tons

Total capacity for the present station, 7,600 tons— six-day supply

Total capacity for the ultimate bunker, 19,000 tons

Each pocket is provided with four air-operated sliding gates

COAL LARRIES

Make	Bergen Point Iron Works
Number and capacity	2 each 25 tons

SLUICING PUMPS

Three 4,400 g.p.m. DeLaval, driven by 175-hp. motor

Drive—175-hp., 1,200-r.p.m., 2,300 v., double-wound-rotor type G.E. motor

METERS AND INSTRUMENTS

Crosby Steam Gage & Valve Co.— Indicating pressure gages for steam, water, air oil, etc.—90

Ashton Valve Co. Illuminated master steam gages— 2

Bailey Meter Co.—Six-point draft gages, 12; four-point draft gages, 12; two-pen recording totalizing steam flow meters, 12

Brown Instrument Co.—Recording flue-gas pyrometers, 16; indicating flue-gas pyrometers, 4

Chas. Englehard, Inc.— Recording thermometers, condenser water and steam, 5; recording thermometers, turbine supply steam, 5; recording thermometers, heaters steam and water, 45; recording thermometers, generator air-cooler water, 5

The Foxboro Co.— Recording pressure gages, 1 pen, turbine supply steam, 10; recording pressure gages, 2 pen, feed water, 1; recording pressure gages, 4 pen, turbine bleeding, 5

Taylor Instrument Co's Indicating thermometers, 207; mercury vacuum gages, 10

Cochrane Corp. — V-notch blow-down meter with leakage recorder, 1

General Electric Co.— Recording and integrating steam-flow meters, 5; recording and integrating flow meters, feed water, 11; recording and integrating flow meters, condensate, 5; recording and integrating flow meters, city water makeup, 5

MISCELLANEOUS EQUIPMENT

Turbine-room cranes	Two installed, 125-ton each, Cleveland Crane & Engineering Co.
Sluice gates	M. H. Treadwell Co.
Revolving screens	Link Belt Co.
Furnace ventilating blocks	Drake Non-Clinkering Furnace Block Co. and Bernitz Furnace Appliance Co.
Firebrick	Henry Maurer and General Refractories Co.
Condenser tubes	British American Metals Co., and the Scovill Mfg. Co.
Smoke flues	Connery & Co.
Strainers	Elliott Co.
Generator air-duct automatic doors	Carrier Engineering Co.
Sump pumps	Buffalo Forge Co.
Refrigeration plant	Brunswick-Kroeschell Co.
Steam-heating vacuum system equipment	Warren Webster & Co., and Nash Engineering Co.
Desuperheaters	Elliott Co.
Ash sluiceways	Allen-Sherman-Hoff Co.

Size, Load and Rating of Boilers

By C. HAROLD BERRY
Associate Editor of *Power*

THE method for expressing the size, load and rating of boilers that has prevailed since the Centennial Exposition in 1876, was devised to establish equality between the rated horsepower of an engine and the boiler horsepower of the boiler furnishing steam to it. This seemed simple and logical, but a fallacy lay in the fact that progress in the economy of engines and in the capacity of boilers was in no way parallel, so that before many years had elapsed each boiler horsepower of boiler size was capable of developing sufficient steam to produce considerably more than one mechanical horsepower of engine output.

In discussing this matter, it is necessary to distinguish accurately three basic quantities, which, curiously enough, are often confused in connection with boilers, whereas in connection with other equipment such confusion would appear ridiculous. These three distinct things are: (1) The *size* of the unit as a physical structure; (2) the *load* on the unit, and (3) the *ratio* of load to the size.

The Case of the Engine

Consider these items for the reciprocating steam engine. The physical size of the engine is fully expressed by giving the diameter of the cylinder and the length of stroke, or the cubic inches of piston displacement (per stroke or revolution). All other dimensions have to do with structural features. The cylinder size gives a definite idea of how big the engine is, but it tells nothing definite about how much power the engine will develop. The power developed will depend not only upon this, but also upon steam pressure, exhaust pressure and speed. Now, of course, one familiar with steam-engine practice would be able to make a fair guess at these quantities for a given size of cylinder, and might therefore estimate roughly the capacity of the engine, but this would be no more than an approximate guess and would have no scientific value.

When the engine is actually operating, its load is measured in terms of the rate of development of mechanical work, in foot-pounds per unit time. Since this would yield large numbers for any actual engine, it is customary to employ a larger unit, the horsepower, which is 33,000 ft.-lb. per min., or 550 per sec.[1]

The ratio of engine load to engine size is a complicated question because of the so-called rated capacity of the engine. Of course the obvious way to express this relation would be in terms of horsepower per cubic inch of piston displacement (for a single stroke or revolution), but because of the characteristics of a steam engine, which are now thoroughly understood, another practice has arisen. If the steam required per unit output is determined for various loads on an engine, it develops that this steam rate is high at very low load, decreases as the load increases, passes through a minimum, and increases with further increase in load. Since the object of an engine is to develop power at minimum cost, it is altogether natural to select the load corresponding to minimum steam consumption as the nominal or rated capacity of the engine. When this is done, any actual load is frequently expressed as a percentage of the rated capacity.

Now if the point of best economy for a steam engine always gave the same horsepower output per cubic inch of piston displacement, it would be possible to determine the rating of an engine by dividing the piston displacement by some sort of constant. However, there is no reason to expect that such a simple relation should exist, and as a matter of fact there is no such easy procedure. The rating of an engine is based on tests made either on the engine itself or on similar engines.

Application to the Boiler

In the case of the boiler a logical method of expressing the size of boiler is to state the square feet of heating surface. It is recognized that this is an unsatisfactory procedure at best, because the location of a square foot of heating surface affects its relative value for heat absorption, but on the whole a simple statement of square feet is perhaps the best that can be done to indicate the size of a boiler.

The load on the boiler should obviously be expressed in terms of what the boiler is intended to do—heat absorption. This can be done approximately by stating the rate at which steam is generated in pounds per hour, but this is not a definite statement, because a pound of steam does not represent a definite quantity of heat. If the feed water is hotter or if the steam pressure or temperature is lower, it takes less heat to make a pound of steam. This was recognized in 1876, and it was then proposed to measure the load on a boiler in terms of the rate of heat absorption. It was further recognized that if this were expressed in B.t.u. per hour, the numbers involved would be very large, and to reduce these to more convenient proportions a large heat unit was proposed, similar to the mechanical horsepower. Just as the mechanical horsepower is by definition 33,000 ft.-lb. per min., so this large heat unit is by definition 33,479 B.t.u. per hour.[2] Unfortunately, the analogy between this new heat unit and the mechanical horsepower was given undue weight, and the new heat unit was called the boiler horsepower.

All this was bad enough, but the next step was worse. It was assumed that a boiler could be rated in a man-

[1] It should be noted in passing that only in the case of energy have units of rate of flow been established. For the flow of water or other fluids compound units are used, such as gallons per minute, or barrels per day, or cubic feet per second. Velocities likewise use compound units, miles per hour, or feet per second. In the case of energy, on the other hand, rate units are common, the horsepower, the watt and the kilowatt. The horsepower is by definition 33,000 ft.-lb. per min., and the kilowatt is by definition 10^{10} dyne-centimeters per sec. Both of these are strictly mechanical units, and the only reason that the kilowatt is used for electrical measurements is that electrical units are defined in terms of the dyne and the centimeter.

[2] The definition of this unit is an odd figure, because it was originally defined as the heat required to generate 30 lb. of steam from feed water at a temperature of 100 deg. F. into steam at a pressure of 70 lb. per sq.in. gage. Later, in recognition of the fact that these limitations did not represent practice in any permanent way, the definition was revised to specify 34.5 lb. of steam from feed water at a temperature of 212 deg. F. into steam at a pressure of one standard atmosphere, giving rise to the familiar expression "from and at 212 deg.," and to the calculation of equivalent evaporation and factor of evaporation, whose usefulness has been the conversion from terms corresponding to actual operation into arbitrarily established artificial terms corresponding to nothing whatever.

ner similar to the rating of an engine. Now it is true that there is a load at which the efficiency of a boiler is a maximum, but there is no fixed relation between this best load and the size of the boiler, because the efficiency of a boiler is greatly affected by the disposition of the surface and for a given boiler the efficiency and best load depend upon the design of the furnace and combustion equipment. It is not far from the truth to say that a boiler has no rating, in the sense that an engine has a rating which expresses more or less accurately the load at which the economy is a maximum.

However, this mistake was made in 1876, and it was assumed that the normal running load on a boiler corresponded to the generation of 1 boiler hp. for each 10 sq.ft. of heating surface. Accordingly, a boiler was rated in terms of horsepower by dividing the heating surface by 10. This gave rise to the confusing situation wherein the term boiler-horsepower means two distinct things, which are not even the same kind of quantity. One is the size of a structure in units of area, each unit equal to 10 sq.ft.; the other, the load or output of a boiler in units of heat flow, each unit equal to 33,479 B.t.u. per hour.

Since the rating of a boiler is expressed in horsepower, and its load under any actual conditions is expressed in the same terms, it is natural to express the load as a percentage of the rating, to indicate the severity of the service to which the boiler is subjected.

Efforts to Clear Up the Confusion

In recent years it has been widely conceded that the situation is entirely illogical and confusing to a high degree, and numerous suggestions have been offered in an attempt to develop a logical scheme.

The method of expressing the size of a boiler is already fairly well cleared up, with practically universal agreement that the proper way to handle this is to state the size of a boiler in square feet of heating surface. The American Boiler Manufacturers Association has agreed to use this unit primarily, and gradually to suppress the use of the term horsepower as a unit of area equal to 10 sq.ft. of heating surface. The American Society of Mechanical Engineers, in the preparation of its Power Test Codes, has relegated boiler horsepower to secondary place. European practice has never developed anything analogous to our boiler horsepower.

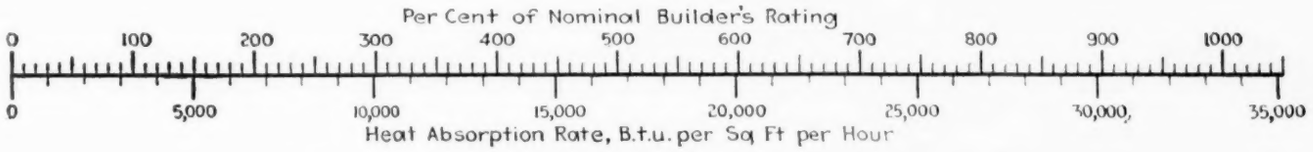
It is now well recognized that the boiler has no definite logical rating, and that, therefore, the only way to express the relation between load and size is to state the heat absorbed per square foot per hour.

The load on a boiler might still be expressed in boiler horsepower, which is a proper quantity defined in terms appropriate for measuring what the boiler is designed to do; that is, it is defined as a definite rate of heat absorption in heat units per unit time. Unfortunately, however, the magnitude of this unit, 33,479 B.t.u. per hour, is an awkward number to handle, and the results calculated with its use do not carry any special significance assisting in the interpretation of boiler results.

The Kilber was suggested some years ago to designate 1,000 lb. of steam per hour, "ki" standing for "kilo," "lb" for "pounds," and "er" for "per hour." This interesting suggestion would be of considerable value in any one plant, where the feed temperature and steam conditions are substantially constant, but for general use it is open to the objection that a pound of steam does not represent a definite quantity of heat absorption by the boiler, and is therefore not a proper unit for measuring the load.

More recently, H. W. Brooks has suggested that the unit be established as 100,000 B.t.u. per hour, and called a "therm."

Still another proposal, indorsed by the A.S.M.E. Power Test Codes Committee, is to use 1,000 B.t.u. as a larger heat unit, and to express boiler load in terms of this unit per hour. They have suggested the name "unit of evaporation" for 1,000 B.t.u. and have proposed the expression of boiler load in units of evaporation per hour, and the relation of load to size in units of evaporation per square foot per hour.



Scale for converting per cent of boiler rating to heat absorption rate

For the present this seems the most satisfactory proposal from all points of view, except perhaps for the objection that "unit of evaporation" is an awkward name. Of course, one can say thousands of B.t.u., but this is just about as awkward. In writing this quantity in abbreviated form, it would be possible to write K.B.t.u., utilizing K as the abbreviation of the Greek Kilo, one thousand, just as in the case of other units, and the unit might be called K.B.t.u., or Kilo-B.t.u.

Mr. Elliott has proposed the use of M.B.t.u., for this quantity, using the ancient abbreviation, M, that has long been common in commercial transactions. This is open to the objection that in the field of heat and electricity most of the technical terms are derived from Greek. The K for Kilo is already in wide use, and M is also widely used as an abbreviation of the Greek Mega, one million, as in the term megawatt.

Since the old per cent of rating, based on the nominal rated horsepower of a boiler, has come to mean much to most boiler operators, the scale for converting from K.B.t.u. per sq.ft. per hour to per cent of the nominal rating, may be of service.

To summarize, the *size* of a boiler should be expressed in square feet of heating surface; the *load* in K.B.t.u. per hour, and the *rating* in K.B.t.u. per sq.ft. per hour.

This scheme is entirely logical and scientific. Each quantity is measured in terms of appropriate units. The load and rating units are convenient, in that in a rough way one K.B.t.u. is approximately equivalent to 1 lb. of steam. Calculations are simplified through the elimination of equivalent evaporation, factor of evaporation and the quantity 33,479. In the absence of some better suggestion it is to be hoped that this system will be adopted until something better turns up.

Keeping Troubles Away From Electric Motors

How Sixty Motors in One Plant Have Been Kept in Operation for Over Twenty Years with Practically no Shutdowns Due to Troubles

By H. R. SAWYER

VOLUMES have been written on the care of electric motors, but in many cases they are not taken care of or receive improper attention. Proper care of motors and controllers will pay handsome dividends in reduced repair bills and improved service, as can be shown by many installations where foresight is practiced instead of hindsight—foresight to detect the little things that lead to trouble and remedy them before they have an opportunity to get in their deadly work of putting the machine out of business.

The first thing to do in the care of electrical equipment is to see that it is properly installed and protected. If a good mechanical job is not done when the motors

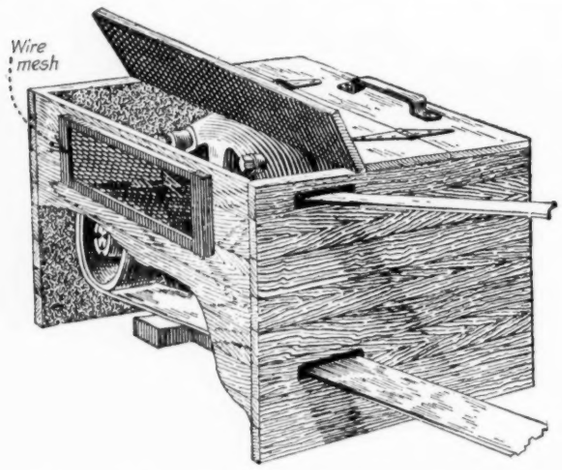


Fig. 1—Motor inclosed in a box for mechanical protection

and controllers are installed trouble is not very far away. Wiring exposed to oil, moisture and dirt or to mechanical injury; motors on poor foundations or improperly lined up; unsuitable driving gear between the motor and the load—all these are harbingers of trouble. If the maintenance man or those to whom he is responsible possess sound judgment, they will remedy all such irregularities when this can be done at a minimum of expense, rather than wait until a breakdown occurs.

Repairing failures in motors and wiring that have been caused by defective installation does not remove the cause of the trouble. This can be done only by making the installation correctly, and it does not cost any more to do this before it has caused a failure in the equipment than after. Therefore, the sensible thing to do is to put the installation in the best condition mechanically for the motor.

Two other things are essential to give the motor a fair show. These are, to give it proper electrical protection and to use sufficiently large conductors so that the voltage at the motor terminals will be maintained near normal. Low voltage on certain classes of motors causes an increase in current to carry a given load, and if the machine is operating at near full load at normal voltage, it may be overloaded at reduced voltage. Low voltage on almost all types of motors will cause reduced speed with a consequent reduction in production. Where motors are driving centrifugal pumps, low voltage may cause the speed to fall so low that the pump will fail to operate properly.

If a motor is not properly protected and trouble does happen, the damage will probably be much more extensive than if proper protection had been provided. Where trouble is experienced with fuses blowing or circuit breakers opening, in most cases the remedy is not a larger fuse or a higher circuit-breaker setting, but the trouble is due to some fault in the electrical installation or the driven load, that should be located and corrected before it causes a failure in the motor or other damage.

There are a number of makes of totally inclosed safety switches on the market. Such equipment should be used on every motor of any consequence, since it provides safety not only to the switch and fuses, but also to the workmen.

Mechanical Protection of Motors

Of equal importance to electrical protection is mechanical protection. Where a motor is exposed to mechanical injury, it should be protected by a substantial box, with opening for air circulation, covered with fine wire mesh. One type of such a box is shown in Fig. 1. How substantial this box should be made will depend on the forces that it may be called upon to resist, but it should be remembered that it is cheaper to provide mechanical protection to a motor than it is to replace or repair it, to say nothing of the cost in lost production caused by having the equipment out of service. Motors should be located conveniently for inspection and maintenance. If installed in places where those who take care of them must risk bodily injury every time they make an inspection or adjustment, they will probably be neglected. When those who are responsible for the installation of the motors do not think enough of them to provide suitable places for them, it is hardly fair to expect the maintenance crew to risk physical injury in their care.

One of the first things to be given consideration in the actual inspection and maintenance of motors is their lubrication. As to the kind of lubricant to use, it is policy to obtain the best grade possible. Proper lubrication is the cheapest kind of maintenance. On sleeve

bearings, a good grade of light lubricating oil should be used. On ball or roller bearings either a good grade of light lubricating oil or a suitable grease may be used. On these bearings the function of a lubricant is not so much to eliminate friction as to protect the balls or rollers and their races from pitting, due to moisture, acid fumes or other corrosive mediums in the atmosphere.

One of the big problems in the proper lubrication of motors, is to keep the lubricant in the bearings and the

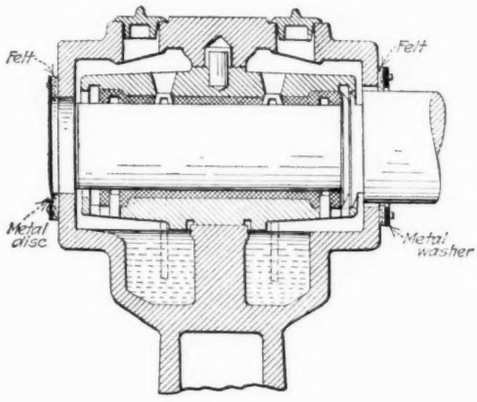


Fig. 2—Method of sealing open-type bearings

dirt out. With ball or roller bearings this is well taken care of in the design. The housings are made practically oil-tight and are so designed that the bearing cannot be flooded and let the oil run into the motor and on the windings.

In the design of the sleeve-type bearing most of the motor manufacturers have been rather remiss, in that proper attention has not been given to making them oil- and dirt-tight. Recently, some of the manufacturers have adopted such types of sleeve bearings, but why this should not have been done in the early development of motors is something for which it is difficult to account.

How Oil Gets Out of the Bearings

With sleeve bearings, the oil rings churn the oil into air bubbles, and when these break a fine spray is formed. If the bearings are open, the air current circulating through the machine's windings also passes through the bearings and in doing so picks up the oil spray and carries it into the windings. Dust and dirt carried in the air will be deposited on the oil-wet surfaces of the bearings. From this it is evident that with open-type bearings conditions are favorable for the oil getting out of the bearings and onto the windings and for dust and dirt getting in. If trouble is to be avoided in the operation of motors, the oil must be kept in the bearings and off the windings.

Investigations have shown that oil-tight bearings will prevent from 50 to 90 per cent of the winding troubles experienced with open sleeve-type bearings. Therefore the bearings should be sealed against oil getting out. Fig. 2 shows a method of doing this that has proved fairly successful. A tight-fitting felt washer is put around the shaft and is held in place by a thin metal washer, which is attached to the bearing housing with screws.

In very dirty places, where the dust contains metallic and abrasive particles, it is advisable to change the oil in the bearings about every three months and at the same time wash them clean with kerosene. The oil taken from the bearings may be filtered and used again in the motor bearings or, as is done in one plant, in the sight-feed oil cups on compressors and in other places where the oil is lost.

There are various types of filters on the market, but where an appropriation is not easily obtainable to purchase one, a satisfactory substitute may be made as shown in Fig. 3. In this particular design the tank, which is made of sheet steel, is welded at the corners. A wire-mesh basket is supported in the top of the tank. A ½-in. felt lining is placed in the basket and a quantity of waste. After the oil passes through this material, it is practically as clean as it was before it was used in the motor bearings. A second wire-mesh screen is placed about halfway down inside the tank, as a guard against dirt getting into the oil compartment when the filter is being assembled.

Amount of Oil in Bearings

Careful attention must be given to the amount of oil put into the bearings and to see that the oil rings revolve freely. When the oil has been brought to the correct height in the bearings, if everything is in proper

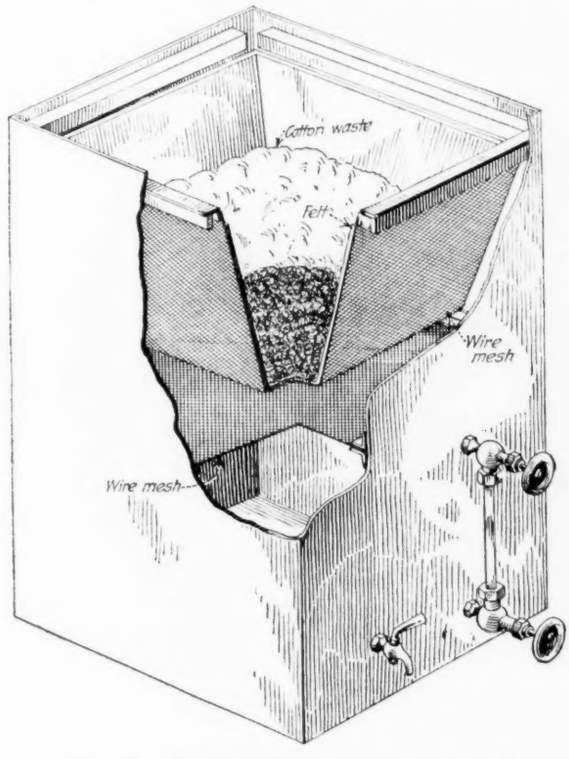


Fig. 3—Home-made filter for motor oil

condition, they should require no more oil for months. It is necessary that the bearings be watched to see that the oil does not leak out and that the oil rings move freely. Too much oil in bearings is about as bad as too little. In the one case the oil works out and onto the windings, which will eventually cause an insulation failure, and in the other, the bearings may burn out and shut the motor down.

Interwoven with lubrication is keeping the motors clean. If oil is allowed to get out of the bearings on the windings, the insulation may be injured, and in addition the oil will cause dirt to stick to the windings

and make it difficult to keep them clean. If the windings are kept dry, one of the best ways of cleaning is to blow them out with compressed air at a pressure of about 40 lb. A compressed-air line may be run to each motor or a small portable compressor can be used for this purpose. If oil gets on the windings and they become gummed up with dirt, about the only way to clean them is with gasoline and a brush, after which they should be thoroughly dried and then given a good coat of baking varnish and baked.

Care of Commutator and Brushes

Where direct-current motors are used, the commutator and brushes must be given careful attention. It is now general practice to undercut the commutator mica. There are a number of devices on the market for doing this, but if one of these is not available, a file may be used, similar to that shown in Fig. 4. When the mica

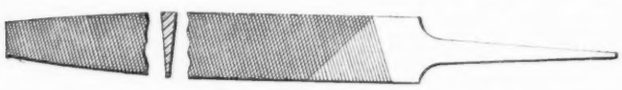


Fig. 4—File used for undercutting commutator mica

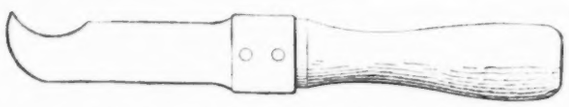


Fig. 5—Tool for cleaning out the commutator slots

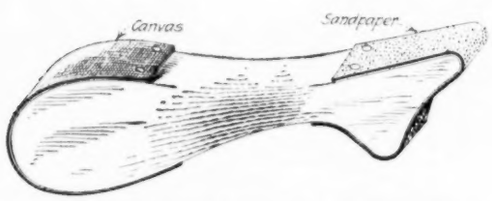


Fig. 6—Tool for cleaning commutators

is undercut, it is necessary that the slots be kept clean. The tool, Fig. 5, made from a piece of hacksaw blade in a wooden handle, is handy for doing this work. Before satisfactory operation at the commutator can be expected, it must have a true surface, such as obtained by turning in a lathe. The finishing of the surface should be done with a commutator stone, as a much truer surface can be obtained with this than with sandpaper.

After the commutator has been put into proper condition generally, the least it is bothered the better, except to see that the surface is kept clean, which can usually be done by wiping with a piece of canvas. The tool, Fig. 6, made from a piece of wood with canvas tacked on one end and No. $\frac{1}{2}$ sandpaper on the other, has been found handy to clean up any slight irregularities in commutator operation. Two or three thicknesses of canvas and sandpaper may be tacked on the ends, and as they become unfit for use, one layer at a time may be removed. The sandpaper surface is slightly curved to approximate the curvature of the commutator.

Selecting Brushes

The question of suitable brushes is open to much discussion. However, one will do well to follow the brush manufacturer's instructions. Many of the manufacturers have comprehensive instructions on the application of their brushes, and these should be carefully studied if trouble is being experienced, or better still, if convenient to do so, consult the brush manufacturer's engineers.

Conditions under which motors operate vary widely, but in the average case if the simple instructions herein given are followed, little serious trouble should be experienced. In one plant that has over 60 motors, ranging in size from $\frac{1}{2}$ to 200 hp., a shutdown of a motor, when it is needed, is practically unknown. Many of the motors have been in operation for over twenty years without changing a bearing. During a considerable part of this period, they have been in operation 24 hours a day. These motors work where there are metallic dust and sand in the air. The chief requisite for keeping out of trouble, has been to study the conditions under which the motors operate, and if any are found that are likely to cause trouble, take the necessary steps to protect the motor in so far as it is possible.

Preventing Heating of Armature Bands

On small and medium-sized armatures it is quite general practice to retain the coils in the slots by steel-wire bands around the outside of the core. After the bands have been put in place, the turns of wire are soldered together to form what amounts to short sections of cylinders around the armature.

What is not generally appreciated by the average electrical worker, is that these bands are conductors on the surface of the armature core, to which the same laws apply as to the conductors in the slots. The section of a band that is under a north pole will have a voltage generated in it in a direction parallel to the armature shaft, when the armature is revolving, and the section under the south pole will have a voltage generated in it in an opposite direction. Since these two sections are both part of the same band, they form a short-circuited coil in which a current can circulate. If the bands are made too wide, the current induced in them may be high enough to cause sufficient heating to melt the solder and release the bands.

Another cause of heating is the use of too many copper clips or the improper location of these clips. The bands are usually insulated from the core with thin mica, and thin strips of copper are placed under and at right angles to the bands as they are wound in place. The ends of these strips are bent up over the bands and soldered in place with the rest of the band. If these copper strips, or clips as they are usually called, are placed one pole space apart, the voltage generated in them will have the same relation as in two sides of the coils, and the current circulating in these clips may be sufficient to produce excessive heating and melt the solder.

In the larger-sized machines it is best to use one clip for each pair of poles, as this allows spacing them so that they will always be under poles of the same polarity and will therefore have a voltage induced in them in the same direction. This will prevent circulation of current from one clip to another through the band. If the bands are made narrow and the clips are properly spaced and used sparingly, there should be no trouble from excessive heating of the bands.

Connecting the bands together electrically should be avoided, since the connectors will have a voltage generated in them the same as in the coils and will increase the current in the bands, consequently the heating will also be increased.

Increasing the Capacity of a Jet Condenser

THE history of four changes made in a jet condenser whereby its steam condensing capacity has been doubled, throws light on the influence of structure on operating characteristics.

By JULIAN HARVEY*

THE Connors Creek plant of the Detroit Edison Co. was one of the first plants to use the house turbine exhausting to an auxiliary condenser where boiler feed water is heated under vacuum[1]. There are five jet condensers in this system, and this article tells the story of the oldest member of the group. The material is summarized in the hope of benefiting those who wish their apparatus would handle "just a little more steam."

This condenser receives as circulating or injection water the condensate from the main turbines and condenses steam exhausted from the auxiliary turbines in the plant. The resulting mixture, being pure distilled water, is used as boiler feed, and the apparatus, because of its dual rôle of condensing exhaust steam and heating boiler feed water, has been christened "heater-condenser." Fundamentally, however, its operation differs in no wise from that of other barometric condensers.

It started innocently enough, with no special promise of future greatness, when the plant was being built in 1914, as an ordinary barometric jet condenser. The specifications read in part:

> Guarantee. Based upon circulating 480 gal. of water per minute and the temperature of the circulating outlet being ten degrees below the temperature of the vacuum, the performance . . . will be as follows:
>
> *With 20,000 lb.* of steam per hour
>
> | | | |
> |---|---|---|
> | 80 deg. Inlet | 159.2 deg. Outlet | 12 in. Abs. pressure |
> | 60 deg. Inlet | 139.2 deg. Outlet | 7.5 in. Abs. pressure |
>
> *With 10,000 lb.* of steam per hour
>
> | | | |
> |---|---|---|
> | 80 deg. Inlet | 119.6 deg. Outlet | 4.5 in. Abs. pressure |
> | 60 deg. Inlet | 99.6 deg. Outlet | 2.6 in. Abs. pressure |

At this point it is necessary to digress momentarily and define "terminal difference," a phrase often entering discussions of condenser or heater performance. In this article it will be used to designate the actual difference between the temperature of the mixture in the condenser tail pipe and the temperature of air-free saturated steam at the absolute pressure prevailing at the condenser steam inlet. Terminal difference is of interest to the operating engineer and may be followed closely by comparing the tail-pipe temperature with the saturation temperature corresponding to the vacuum gage reading as found from a curve posted in the condenser room, such as that of Fig. 1. This curve is based on a barometer reading of 29.7 in. of mercury, and will be somewhat different for other conditions of atmospheric pressure.

A terminal difference of more than ten degrees will usually indicate: (1) air entering in excessive quantities with the steam or through leaks; (2) fault in the dry vacuum pump or its equivalent, or (3) approach of the steam quantity to the maximum capacity of the condenser.

Returning to No. 1 barometer condenser, it appeared on the scene in due time, looking like Fig. 2. As may be seen, the injection water was delivered through the side of the condenser to a tray at the top, from which it spilled in a series of sheets or sprays. The steam entered approximately tangentially and followed a circular passage which constantly reduced in cross-section and which terminated in the suction of the dry vacuum pump. The mixture of condensed steam and circulating water dropped down through the tail pipe.

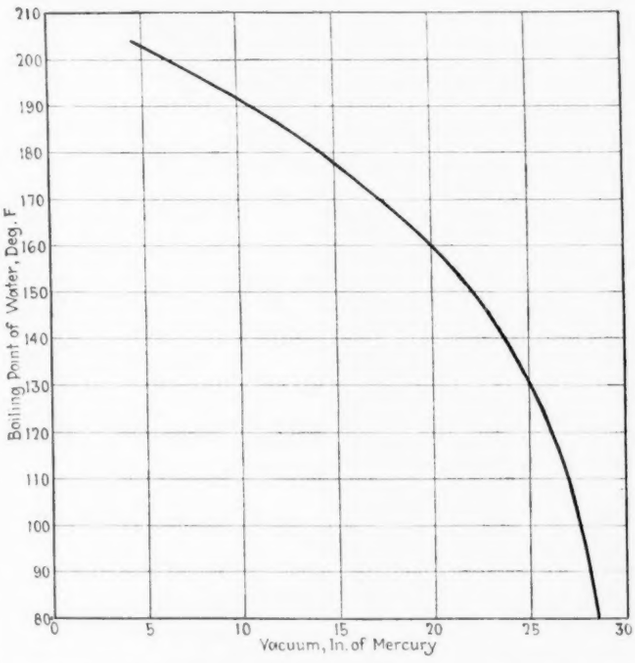


Fig. 1—The relation between vacuum and boiling point for a barometer reading of 29.7 in. of mercury

During February, 1916, observations were taken of the performance of the condenser under various conditions of steam and water flow. Four readings have been selected from this test, which covered a wide variation of vacuum and steam quantities. These readings have been arranged in order of increasing vacuum and are given in Table I.

It will be observed that as the circulating water rate increased, the rate of steam condensation decreased and the terminal difference increased. This was probably due to the fact that at higher rates of flow the water spilled over the trays in a sheet which did not have

*Formerly technical engineer of the Connors Creek plant of The Detroit Edison Co. Mr. Harvey wrote this article shortly before his death on June 30, 1926.

[1]For more complete discussions of the system, see *Power* for Sept. 14, 1915, page 388, "Transactions, A.S.M.E.," Vol. 37 (1915) p. 1085, and Vol. 43 (1921) p. 500.

sufficient time to break up into spray before reaching the tail pipe.

In the hope of improving the performance of the condenser, the height was increased to about double its original value by adding a cylindrical piece three feet high to the top, and raising the water tray a corresponding amount. This arrangement is shown in Fig. 3. Readings taken under these conditions were in part as given in Table II.

TABLE I—ORIGINAL CONDENSER PERFORMANCE, CONDENSER AS IN FIG. 2

Vacuum in Condenser, In. of Mercury	Abs. Pr. in Condenser, In. of Mercury	Steam Condensed, Lb. per Hour	Tail-Pipe Temp., Deg. F.	Circ. Water Temp., Deg. F.	Terminal Diff., Deg. F.	Circ. Water Gal. per Min.	Heat Transfer, Millions of B.t.u. per Hr.
10.5	19.1	22,200	189	52	1	258	17.66
15.6	14.0	19,600	172	52	4	207	12.40
21.0	8.6	13,320	142	52	13	242	10.90
24.5	5.1	11,530	105	53	45	384	10.10

The increase in volume having no significant effect on condenser performance, the next step was to improve the arrangements for withdrawing air. This was done by adding a second filler, this time one foot in height, at the top of the condenser and providing an additional suction for the dry vacuum pump, in the top or head of the condenser. Fig. 4 shows this arrangement. At this time also a splash tray was placed in the cone at the bottom of the condenser, with a view to preventing uncondensed steam from being carried down into the tail pipe. Tests made following this change showed a decided improvement in terminal difference, which was held at or below ten degrees throughout a wide range of vacuum. It was observed, however, that water was sometimes drawn into the air-pump suction in the condenser head despite the baffle at this point. No harm was done to the pump, as the piping had been arranged with this contingency in mind.

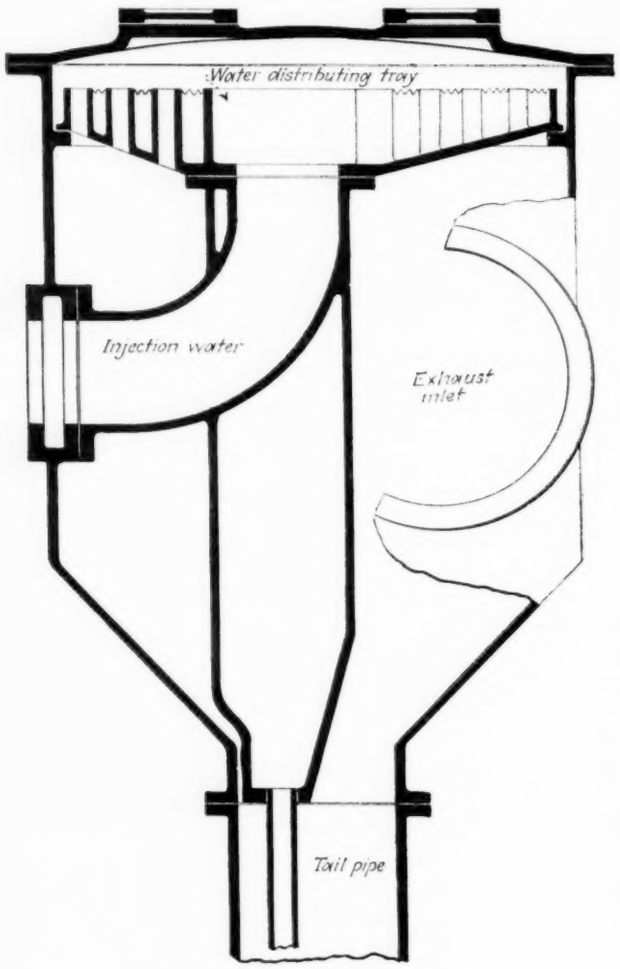


Fig. 2—The original condenser

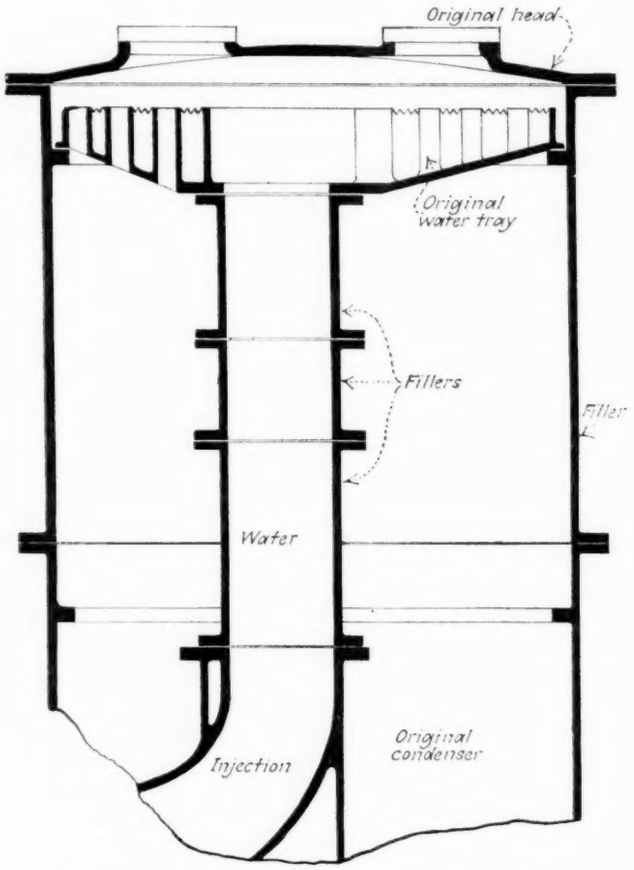


Fig. 3—The first change was the addition of a 3-ft. filler in the shell

No further tests were made for several years, and during this interval the only change in the apparatus was the replacement of the original circulating pump by a larger one of 1,200 gal. per min. capacity. In 1919 it was found that the maximum capacity of the condenser was 28,000 lb. of steam per hour, with a circulating water flow of 363,000 lb. per hour at an absolute pressure of 14.8 in. of mercury. Increase of water flow beyond this point was accompanied by a decrease of steam condensation, probably due to the same phenomenon as previously noted. It was also observed at this time that the condenser would reach a condition of instability when approaching its maximum capacity. The instability was manifested in a cycle of events that were to some extent possible only due to the particular arrangement of

TABLE II—PERFORMANCE OF THE CONDENSER SHOWN IN FIG. 3

Vacuum in Condenser, In. of Mercury	Abs. Pr. in Condenser, In. of Mercury	Steam Condensed, Lb. per Hour	Tail-Pipe Temp., Deg. F.	Circ. Water Temp., Deg. F.	Terminal Diff., Deg. F.	Circ. Water Gal. per Min.	Heat Transfer, Millions of B.t.u. per Hr.
10.4	18.7	18,600	181	68	8	260	14.7
14.5	15.3	13,900	153	55	17	260	12.8
19.2	10.6	13,300	131	55	33	320	12.2
21.2	9.6	13,400	119	56	41	378	11.9

apparatus in the plant. Without going into great detail it may be said that the instability was due to the sensitiveness of the circulating pump to slight changes in vacuum in the condenser when at or near maximum capacity. Inasmuch as the total head against which the pump worked was little more than the absolute pressure in the condenser plus the lift (about fifty feet),

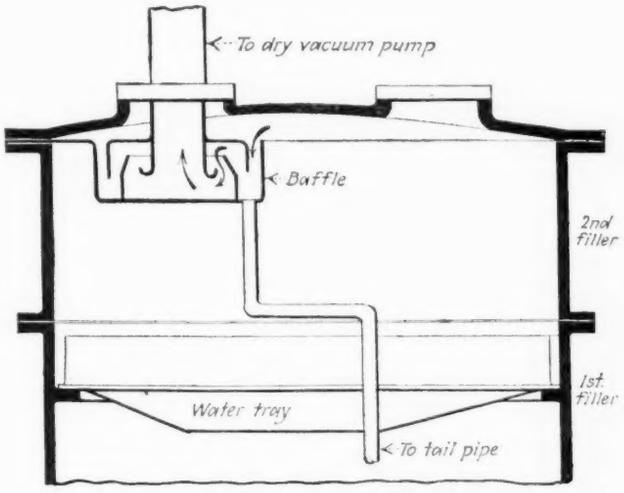


Fig. 4—The second change added a new filler and a new air-pump suction

a slight drop in vacuum in the condenser resulted in a greater head on the pump and a decrease in water flow. Decreased water flow resulted in a further decrease in vacuum, causing a progressive rise in absolute pressure. Owing to local conditions the auxiliaries exhausting to the condenser finally unloaded partly, due to the lower exhaust vacuum, and this unloading sufficiently reduced the steam flow to the condenser to enable it to increase the vacuum to its original value. The auxiliaries picked up load once more under the favorable conditions, and the resulting increase in steam flow set the stage for a repetition of the cycle. It was decided at that time that the introduction of a restriction of suitable nature,

TABLE III—PERFORMANCE OF THE CONDENSER SHOWN IN FIG. 5 WITH SPLASH ANGLES REMOVED

Vacuum in Condenser, In. of Mercury	Abs Pr. in Condenser, In. of Mercury	Steam Condensed, Lb. per Hour	Tail-Pipe Temp., Deg. F.	Circ. Water Temp., Deg.F.	Terminal Diff., Deg.F.	Circ. Water, Gal. per Min.	Heat Transfer Millions of B.t.u. per Hr.
15.4	13.2	22,500	172	67	1	439	21.3
15.0	13.6	30,000	166	70	9	644	30.9
9.0	19.5	41,900	180	65	11	756	43.5

increasing the total head on the pump, would make it less sensitive to changes in condenser pressure and eliminate the instability. Meanwhile attention was diverted to some of the other condensers, where experiments were made with spray nozzles, which might, it was felt, meet the need for resistance in the water line as well as provide good distribution of condensing water.

Finally, in 1925, No. 1 condenser was once more revised, as shown in Fig. 5. As may be seen, a single suction for the dry vacuum pump was provided at the top, although a new baffle was installed in an effort to prevent entrance of water into this line. The steam entrance was left as before—near the bottom and substantially tangential. All other "internals" were removed except the splash tray at the bottom, and replaced by 60 spray nozzles mounted on a set of radiating pipes (like the spokes of a wheel) near the top, with suitable piping to supply them with water. Just above the steam inlet a series of five angle splash trays was installed to insure the complete breaking up of any streams of water that might form.

Tests on the condenser as rebuilt indicated a slight improvement in capacity over the arrangement immediately preceding, but showed the same characteristic of a decrease in steam condensation when the water flow was raised above a certain point—in this case, about 600 gal. per min. The theory was finally evolved that the splash trays were at fault—that the openings were not sufficient to allow free passage of steam and air upward while the water was flowing downward, and reduced the effective volume of the condenser to the

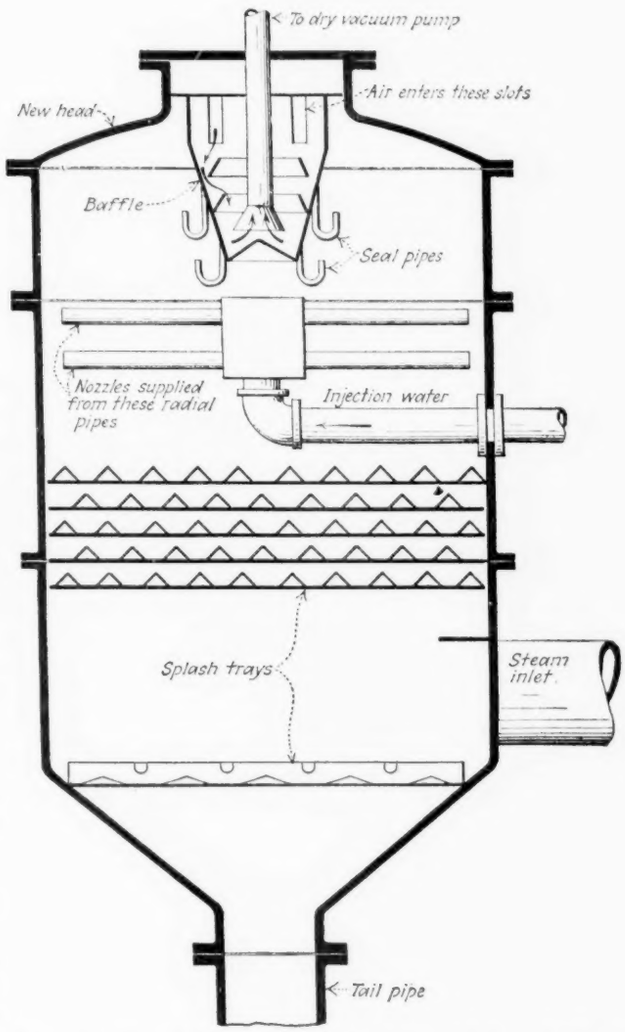


Fig. 5—The third change was a complete rebuilding with spray nozzles instead of the top water tray

space below the trays. Acting on this theory the splash trays were removed entirely in the spring of 1926. No other changes were made.

Recently tested, the condenser was found to have almost doubled its original capacity, notwithstanding the fact that the circulating water was somewhat warmer than during the first test. Three points from the latest test are given in Table III.

From the work done on this condenser some general conclusions may be drawn as to design characteristics

necessary for most efficient operation. These may be enumerated briefly as follows:

1. In general, steam should be introduced at the bottom of the condenser and condensing water at the top.

2. Air should be drawn off above the water inlet, so as to afford the maximum possible cooling effect and consequent reduction in air volume. Suitable baffles, of course, should be provided to prevent water entering the vacuum pump suction line. As a final protection for

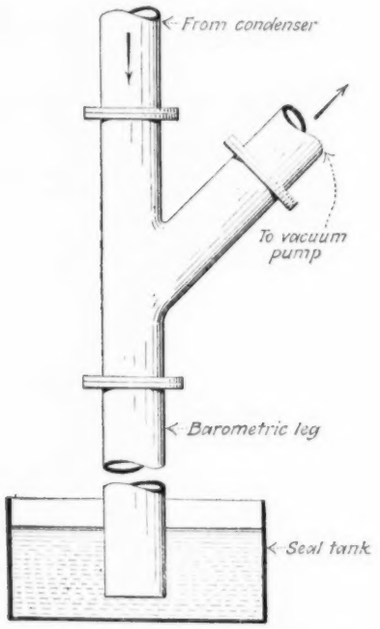


Fig. 6—Trap in air-pump suction line will keep water from entering the pump cylinder

this pump, should any be necessary, a lateral may be placed in the suction line in a vertical position, as shown in Fig. 6.

3. Provision should be made to form a spray of the circulating water. In general, nozzles for this purpose require less space than trays, and therefore permit more capacity in a given sized condenser. Sometimes an existing installation of nozzles may be increased in capacity by enlarging the holes. Care should be taken in making this change to make sure that the holes are not enlarged to the point where the spray degenerates into a jet or stream.

4. A resistance of some appreciable amount between the circulating pump and the condenser makes the rate of water flow less sensitive to slight changes in absolute pressure in the condenser. Spray nozzles seem to meet this requirement satisfactorily.

In conclusion it may be said that this article is not intended to furnish information whereby any jet condenser may be radically improved without the expenditure of time or money. The experience here set down may, however, point the way to some changes in the direction of increased capacity or vacuum without the necessity of adding another complete condenser with all auxiliaries.

Naturally, a condenser working alone on a turbine or engine exhaust will behave differently from one of a group receiving steam from a common header. The peculiarities of this arrangement may explain some apparent inconsistencies in the data.

Increase in Power Per Wage Earner

SIGNIFICANT in American industrial development during the last decade has been the relatively greater increase of production over that of wage earners, indicating a striking saving in labor, principally in manual and common labor. This, in the light of studies made by the National Industrial Conference Board, is accounted for to a large degree by the greatly extended use of power in manufacturing operations. These studies have been based largely upon data collected by the U. S. Census Bureau.

From 1914 to 1923, the period for which data are graphically presented in the accompanying chart, wage earners in all manufacturing industries increased from 7,023,685 to 8,778,156, or less than 25 per cent, while production, in terms of value added by manufacture, increased from $9,857,580,000 to $25,850,300,000 In terms of dollars of 1914 purchasing value, eliminating the change in the price level so as to obtain an indication of the volume of production represented by these figures, the increase ranges from $9,857,580,000 to $16,497,000,000, or more than 67 per cent, as against the 25 per cent increase in the number of wage earners. The reason for the relatively larger production, according to the Board, is to be found primarily in the increase in installed primary power from 22,401,401 hp. in 1914 to 33,094,228 hp. in 1923, a growth of 47.7 per cent, but also in more efficient organization and operation.

No Figures Available for 1921

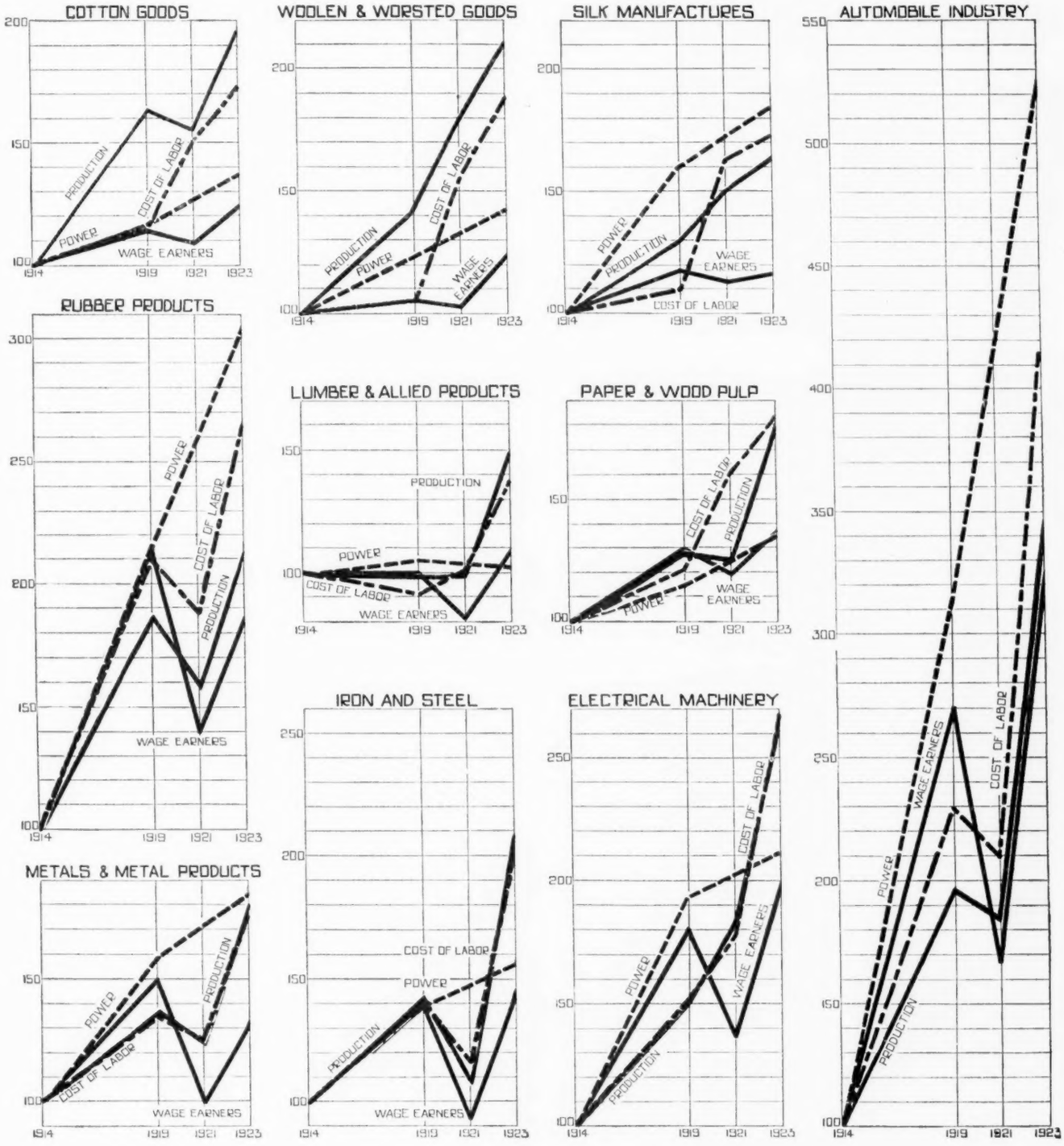
In studying the curve of installed primary power applied in the various industries given in the chart, it should be kept in mind that no figures on installed power are available for the year 1921, no census of power in manufacturing industries having been taken in that year. While the power curve for each industry indicates the net increase from 1914 to 1923, its intersection with the line marking the year 1921 in many instances probably would be higher than indicated in the chart, and in some instances may possibly have been even higher in 1921 than in 1923, for it must be remembered that much plant equipment was installed in 1920, the "inflation year," in anticipation of greatly increased business the following year.

Although these expectations met with great disappointment in 1921, one of the most critical "deflation years" in American industrial history, it is noteworthy that the increase of power installed during the post-war inflation period in many industries was not only absorbed, but even outstripped, by the demands of natural industrial development during the next two years, as indicated by the great net increase in power and production in various industries by 1923. Installed power being not the same as applied power, but necessarily always somewhat in excess of the total amount of power actually used, it is significant that there was actually nearly 20 per cent less power installed per unit of production in 1923 than there was in 1919, indicating the greater application of power equipment with increasing productive activity and, therewith, the absorption of surplus equipment that had been installed during the inflation period.

Notable is the relative decrease in the number of

wage earners in proportion to production from 1921 to 1923 in nearly every industry represented in the accompanying chart, and the influence of the concomitant increase in power reflected thereby in almost every instance is distinctly traceable. From 1914 to 1923 installed power as well as production increased more than the number of wage earners in every industry studied excepting lumber and allied products, and the paper and wood pulp industry.

Each of the several industries here charted reflects the marked depression in the "deflation year" 1921, when, with the drop in sales and dwindling prices, many manufacturers were forced to sell at a loss in their endeavor to absorb part of their overhead and hold their organizations together, while working forces were severely curtailed. By 1923, however, the curves were pointing upward again at nearly the same rate as before.



Growth of Power in Industry

These charts are reproduced from a study by the National Industrial Conference Board, entitled "Elements of Manufacture." Census figures formed the basis of the study. In the group of industries shown the horsepower used shows a steady increase, the ordinates representing index figures in per cent. The horsepower per wage earner has also increased

Building Up an Organization To Operate the New Plant*

YOU have been appointed to the superintendency of the new power station of the P. D. Q. L. & P. Co., a station of xyz capacity. You are the first operating man on the job, or should be. It is up to you to build up an organization of one hundred or one hundred and fifty men, more or less, according to the size of the plant and the kind of fuel to be used.

You are going to hire these men yourself, so as to be sure that you get just the kind and type of man that is needed for each job, or as near that as is humanly possible. It is a task that will demand your undivided attention.

A foreman or head (you may call them what you will) will be needed for each department, also an assistant foreman for each department if the plant is large enough to warrant it. These foremen should be specialists in their particular lines. You will need a turbine-room man, a boiler-room man, an electrical foreman, a yard foreman, a machine-shop foreman, a man to have charge of the stockroom and tools, possibly a chemist, as well as the assistant foremen, engineers, oilers, pump men, turbine or engine repairmen, a rigger, switchboard operators and assistants, water tenders, firemen and helpers, ashmen, conveyor men, coal-crusher man, boiler-repair men, stoker-repair men, pipefitters and helpers, two or three handy men, derrick engineers and firemen for the yard, two or three good machinists for the machine shop, a blacksmith's helper, stock clerk, office force—and a good *male* stenographer.

THESE are no armchair theories. The author, who is now chief of a large station, knows power plants; they have been his life work. He knows organizations, because he has built them and lived with them after they were built. Above all he understands men. If you face the problem of building an operating organization or expect to be in that position some day, don't pass up this chance to profit by another man's long experience!

Another mighty handy man to have is an inspector, preferably one with some technical training—this man to report direct to you. I don't mean an old sleuth, but a man to check reports in person, to make periodical inspections of everything about the plant—in short, a man to relieve you of a lot of the details, so that you will have more time to attend to the bigger problems that come up every day.

Making up your organization is one of the biggest and most important jobs you will ever be called upon to do, and probably the most important as far as this plant is concerned. Your future success and peace of mind depend on the care you exercise in making up the personnel. The future of the station is also at stake. It will make the difference between poor and high economy, the difference between the apparatus being ready for service 100 per cent of the time and some of the time, the difference between a happy, loyal corps of willing workers and a lot of dissatisfied, discontented bolsheviks, ready to knife you or the plant at the first opportunity. You may be captain, but the crew can sink the ship.

These are the things you want to keep in mind—not only keep in mind but act on; there is no use in keeping them in mind if you don't carry them out. Don't think, "Oh, well, I'll put him on today and fire him when I get a better man." Don't do it! You only think you will fire him at some subsequent time. If it were just one man, it would not be so bad, but if you transgress in one instance you will soon be doing it in many others, and the first thing you know you will have an organization made up of God knows what.

So much for that. Now let us start in by hiring your supervisory force. Right here is where you want to "stop, look and listen," as well as think. Use your old or young think tank as you never did before; you are now hiring the men who are to carry out your orders, the men who are to make up your staff, the men who will have charge of the turbine room, the boiler room, the yard, etc. You will need one type of man for one place, and a different type for another, but they must all be on the level, square, of good clean character, and *absolutely* dependable twenty-four hours a day when the need arises, and know how to be boss of their respective departments without being too hard-boiled or too soft.

In addition to the foregoing the turbine-room man must know turbines, engines, pumps, etc. The boiler-room man must know boilers, fires, piping, traps and so on. If you have a combustion man—and you should have—so much the better; the boiler-room duties can be divided between the two. But one of them must be in charge, unless they both report to you. In the latter case you will have to draw a line between the two, so that there will be no conflict of authority. In fact this must be done in the case of all the different departments. Each man must understand just where to start and where to let go. If no line of this kind is drawn, there is bound to be friction sooner or later.

The yard foreman must know how to pile coal to the best advantage and how to handle cranes, derricks and locomotives. He should be able to speak a little Italian or Polish. If he cannot speak these languages, he should have a right bower who can. In any event he should be able to speak English so that his men will know what he is talking about, and have the patience to explain to them what they are to do. One of the great causes of a lot of trouble in handling foreigners is that they do not understand English, and consequently don't get their orders straight. They do what they think they are told; it may be right, and it may not. You need a

*For personal reasons the name of the author is withheld.

yard foreman with patience and tact enough to see that his men understand what he is talking about.

The electrical foreman should have some technical training, be familiar with generators, motors, oil switches, transformers, wiring switchboards, etc. He should be quick of step, cool, deliberate, careful and above all sure of himself, knowing every move to make and just what to do next and when to do it.

The stock clerk must meet a great many requirements to be successful. He must know his stock, having parts for the turbines, pumps and all auxiliaries, parts for boilers, stokers, pipe fittings, etc., spare parts for the electrical equipment, wire of all kinds and sizes, parts for the derricks, and a thousand and one other things, besides the various supplies such as packing, waste, oils, tools, tape, wire, etc. He should be able and thorough in keeping records and know the location of everything carried in the stockroom, as well as the price, who sells it, and where things not carried in stock can be procured in a hurry when needed.

A machine-shop foreman must first of all be a high-class mechanic. As a rule he and his men are about the only real mechanics around a power station, so you may have to sacrifice some of the aforementioned requirements to get a thorough all-around machinist, but don't do it except as a last resort. Good all-around mechanics are like good stock clerks, hard to find. A good carpenter is another mighty handy man to have, particularly if you can get one who will do other work when not carpentering, for there is not enough work in his line to keep him busy much of the time.

Then there are the assistant foremen; in a plant of any size there should be an assistant to the head of each department. As much care should be used in picking these men as their superiors, except that they should if possible be younger, and of course they will not have had the same amount of experience as the older men. They will get this on the job.

In making up the rank and file of the organization, always keep in mind that some day these men, or some of them, will be promoted to the various foremen's jobs, so stick to the foregoing qualifications as closely as possible—right down the scale to the toolroom boy.

Avoid an agitator as you would a leper; one or two men of this kind can soon spoil any organization. If by any chance you have gotten any of this type in your crew, get rid of them at the first opportunity.

Where are you going to get men with all these virtues? Hunt for them; this glorious country of ours is full of good men. They may not be just the kind of men you are looking for, as far as their experience in power-station work is concerned, but if they meet most of the other requirements, it is a simple matter to train them in your own way. They will be better than men trained somewhere else, with a lot of old tricks to unlearn.

Another thing to be avoided is too many men of one type. For example, don't have too many above forty years of age (to start with) nor too many below twenty-five. You need young men to carry on and older ones to act as balance wheels. Above all don't have too many of one nationality, or of the same fraternity, too many brothers or cousins; mix them up.

As soon as all the supervisory force has been hired and are on the job, have them come together in your office, so that you can tell them just what is expected of each, where his work begins and ends. As they are all present, there will be no misunderstanding among them if you make your statements clear. There may, of course, be a number of places where one man's work will overlap or conflict with another's, and it is very important to straighten out this condition as soon as possible. The time to do this is while they are there together. It is well to have the stenographer present at this conference, so that a complete record of such instructions may be given to each man for future reference, as well as copies for the office files. If necessary, the work should be gone over several times until everything is laid out to the best advantage. If you find that it is better to make changes, go ahead; but be sure to have the heads concerned understand what the changes are.

There are other important matters to take up at these conferences: How do you want the rank and file handled or treated? Who is to be in authority when they (the foremen) are away? Who has charge of each watch or shift? How about days off, overtime, inspections,

Making up your organization is one of the most important jobs you will ever be called upon to do

reporting trouble, making and keeping records or logs, the time of day the shifts change, the time the men are to be on the floor before taking their watch and a thousand and one other details that should be decided at the start to avoid future trouble? All these matters should be settled and put in writing.

Continuity of service is the most important thing in a power station. This fact should be impressed on all hands in such a manner that they will never forget it for one instant. A station that cannot be relied upon will soon lose out. You must give reliable service twenty-four hours a day, every day in the year, keeping the voltage and frequency normal at all times. You cannot do this unless you have an organization that is thoroughly reliable and trustworthy from top to bottom.

If you are a cool, level-headed man, with a lot of good common sense (and you must have if you expect to run a power station), you will soon have a corps of men with the same sterling qualities. A rattle head will soon spoil the best organization in the world.

Next to continuity of service, efficiency is the thing to work for. If you have been careful in making up your organization and have a corps of loyal, willing and conscientious men properly trained and instructed in their various duties, there should be very little trouble in giving continuous service and maintaining the highest economy of which the plant is capable.

In the Diesel Plant

Operating Costs Outweigh Capital Charges

By L. H. MORRISON

THE POINT has often been raised that the oil-engine plant entails too much investment, which by reason of the high overhead charges, partly if not wholly wipes out the operating saving compared to other kinds of power units. However, overhead charges, regardless of how calculated, need not be the determining factor in an industrial power plant, for there are other influences susceptible to control that are of equal or greater weight in arriving at the total cost of power.

This statement may appear to go contrary to current opinion but it rests on a sound basis. In fact, it is possible, by proper attention to operating details, to lower the operating cost so that the capital investment could have been doubled without an increase in the actual total power costs. In other words, capital charges may be of less significance than are the operating costs.

Cost of Power

These facts were recently brought out in a study of a textile plant where Diesel engines were under consideration. This plant purchased 812,000 kw.-hr. of electrical energy per month, of which 129,000 kw.-hr. was consumed at night. The peak day load was 3,900 kw. and at night was 450 kw. The connected load was 5,500 hp., which, at 50 cents per horsepower per month, gave a demand charge of $2,750. The electrical rate was 0.9c. per kw.-hr. for day service and 0.65c. for service between 5 p.m. and 7 a.m. The coal charge was 0.262c. per kw.-hr. These rates brought the total cost per month to $11,862.94, or 1.46c. per kw.-hr.

In a study of the plant load with the idea of installing Diesel units, it seemed desirable to use two 2,000-kw. units to carry the day load and a 500-kw. unit for the night load. This seemed more desirable than to install three large units and run one of these at night. The fuel consumption would be higher, as would also the capital investment.

A tabulation of the operating costs was made, oil being assumed to cost $1.68, $2, $3 and $4 per barrel. The high load factor would result in an economical fuel consumption, and it was assumed that at least 450 kw.-hr. could be produced from a barrel of oil. The lubricating-oil consumption was taken as one gallon per 2,000 kw.-hr. output. The maintenance or repair cost was set at both 1 and 2 per cent of the engine investment. It seemed that a chief engineer and two men, one of whom would be experienced, could easily handle the plant during the day, while one engineer and one oiler would be on the night shift. To span the time between 5 p.m. and 8 a.m., it was planned to set the hours of the day men so that one could work until 7 p.m. and one could start at 6 a.m. A tabulation of the several items gave the following results for the cost of generating 812,000 kw.-hr. per month:

Price of oil per bbl.	$1.68	$2	$3
Fuel, 450 kw.-hr. per bbl.	3,030	3,610	5,410
Lubricating oil	300	300	300
Repairs	375	375	750
Supplies	100	100	100
Labor	795	795	1,030
Monthly operating costs	$4,600	$5,180	$7,590
Overhead (10 per cent per year on $450,000)	$3,750	$3,750	$3,750
Total	$8,350	$8,930	$11,340
Cost per kw.-hr., cents	1.03	1.10	1.40
Overhead (10 per cent on $585,000)	4,875	4,875	4,875
Total	$9,475	$10,055	$12,465

The advantage of using engines having low fuel consumption can be seen from a study of the monthly fuel cost. If the units delivered only 400 kw.-hr. per bbl., as is the case with many inferior designs, the fuel item, with $3 oil, would be $6,090 per month. On the basis of 10 per cent overhead the more economical units would be justified at an additional cost of $81,600. One should also remember that as a general rule the economical engine is the most reliable, for the better efficiency denotes less carbon formation in the cylinders.

FUEL OIL

Pittsburgh—June 29, f.o.b. local refinery; 30@34 deg., fuel oil, 6¾c. per gal.; 36@40 deg., fuel oil, 7½c. per gal.

Dallas—July 3, f.o.b. local refinery 26@30 deg., $1.70 per bbl.

Philadelphia—July 2, 27@30 deg., $2.62½@$2.68½ per bbl.; 18@22 deg., $1.93@$1.99; 13@19 deg., $1.75@$1.81 per bbl.

Cincinnati—June 29, tank-car lots f.o.b. local refinery, 24@26 deg. Baumé, 6½c. per gal.; 26@30 deg., 6⅝c. per gal.; 30@32 deg., 7c. per gal.

Chicago—June 28, tank-car lots f.o.b. Oklahoma, freight to Chicago, 92c. per bbl.; 24@26 deg., $1.25 per bbl.; 26@30 deg., $1.30; 30@32 deg., $1.45.

Boston—July 6, tank-car lots, f.o.b. 12@14 deg. Baumé, 4.42c. per gal.; 28@32 deg., 5.8c. per gal.

Fig. 1—Comparative prices of fuel oils

Examination of the tabulation shows that the purchase of oil at $2 per bbl. in place of $3 oil would save $1,800 per month when using one barrel per 450 kw.-hr. If one assumes 10 per cent as covering actual interest and depreciation, the saving by the use of the cheaper oil would justify an additional investment of $216,000 or almost $50 per kilowatt. If an engine was obtained that could make use of oil selling at 4c. per

gal., this being the heavier grade of boiler oil, the savings over $3 oil would justify an additional investment of $285,500. If, as is frequently the case, the purchaser desires not only to make the interest and depreciation charges, but also 10 per cent profit as well, the permissible additional initial investment to accommodate the use of the cheap fuel would be $142,750, or approximately $32 per kilowatt. Inasmuch as there are many plants using a fuel oil of 30 deg. Bé. gravity and above, costing $3 per bbl., evidently managements are not obtaining the economy possible, or they have been so unwise as to install inferior engines. Viewed from another angle, this possible fuel saving would return approximately 4 per cent on the entire investment, if the cost of the engines remained unchanged.

The point has frequently been made that cheaper oils are not justified, since valves must be reground more frequently. As long as no adjustments are needed, there is little use for the engineer and oiler. These men's duties consist essentially of keeping the machine in working condition, consequently occasional regrinding of the valves, etc., should be no more objectionable than the cleaning of boiler tubes, and the periods between such operations would never be less than 60 days, regardless of the kind of fuel used. Furthermore, dirty oil can be cleansed by putting it through a centrifugal, which will remove the dirt and water.

The plan of burning the cheaper grades of oil applies to small as well as large plants. In a New York pumping plant a 200-hp. Diesel consumes 160 gal. of 9-cent kerosene daily, the yearly cost being $4,350. Oil is procurable at 5c. per gal. which, if used, would yield a yearly saving of over $1,700. As this is 8 per cent of the cost of the engine, the possible fuel saving would take care of the normal overhead charges.

If a plant is operating 24 hours a day for 7 days a week, the use of very cheap oil might, as a precautionary measure, dictate the installation of a stand-by unit. Under such operating conditions the cost of the stand-by would largely if not wholly eliminate the saving obtained through the cheaper oil.

The cost of lubricants shown in the tabulation is a conservative one, although some designs use only half this amount and some require five times the amounts set down. Many cases are known where not over 400 kw.-hr. is generated per gallon of lubricating oil. Such engines in the case under consideration would entail a consumption of more than 2,000 gal. a month, costing over $1,000. As this represents an excess of $8,000 a year, one of the points to be investigated upon the purchase of an oil engine is the guaranteed lubricating-oil consumption, and this guarantee should extend beyond the short shop test.

There is a general belief that stand-by units are necessary. If a plant's units are so sized that two-thirds of the load can be carried with one unit down, the expense of a stand-by unit is not always justified. In the plant under discussion the installation of three 2,000-hp. units and the elimination of the 500-kw. night-load unit would entail an additional investment of $150,000, making an additional monthly overhead of $1,250. The running of one of these units at night on a 300- to 400-kw. load would cause a considerable increase in the fuel consumption.

It is true that trouble with one engine would cause the mill's output to be curtailed, but the chances of such an occurrence in a 50-hour-week plant is small. As a further safeguard the management should take out insurance. For example, let it be assumed that the company's gross profit is $3,000 a day and that a shutdown would entail a loss of $1,000 of this. It would be feasible to insure the plant against shutdown for about $350 per year. When there is no stand-by unit this type of insurance goes into effect on the eighth day after a shutdown. The mill then would lose $1,000 a day for each shutdown of 8 days' or less duration. On the assumption that a stand-by 2,000-kw. unit would cost $200,000 and that interest and depreciation of 10 per cent and profit of 10 per cent are desired, this entails a yearly cost of $40,000, or engine trouble could cause shutdowns for 40 days per year without loss

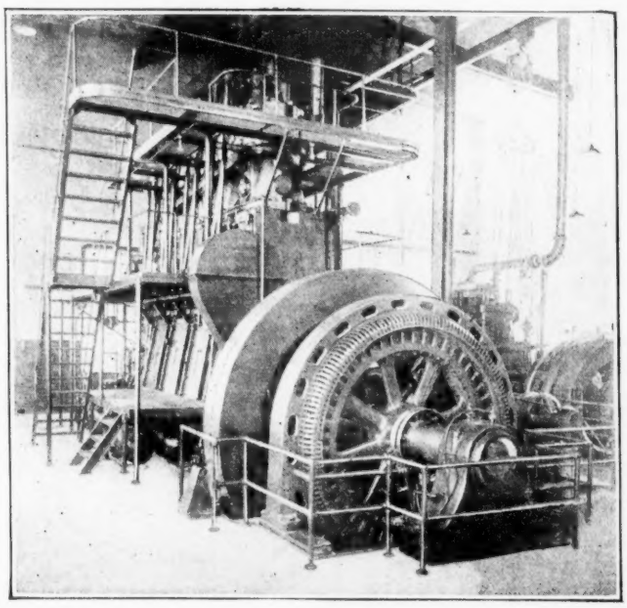


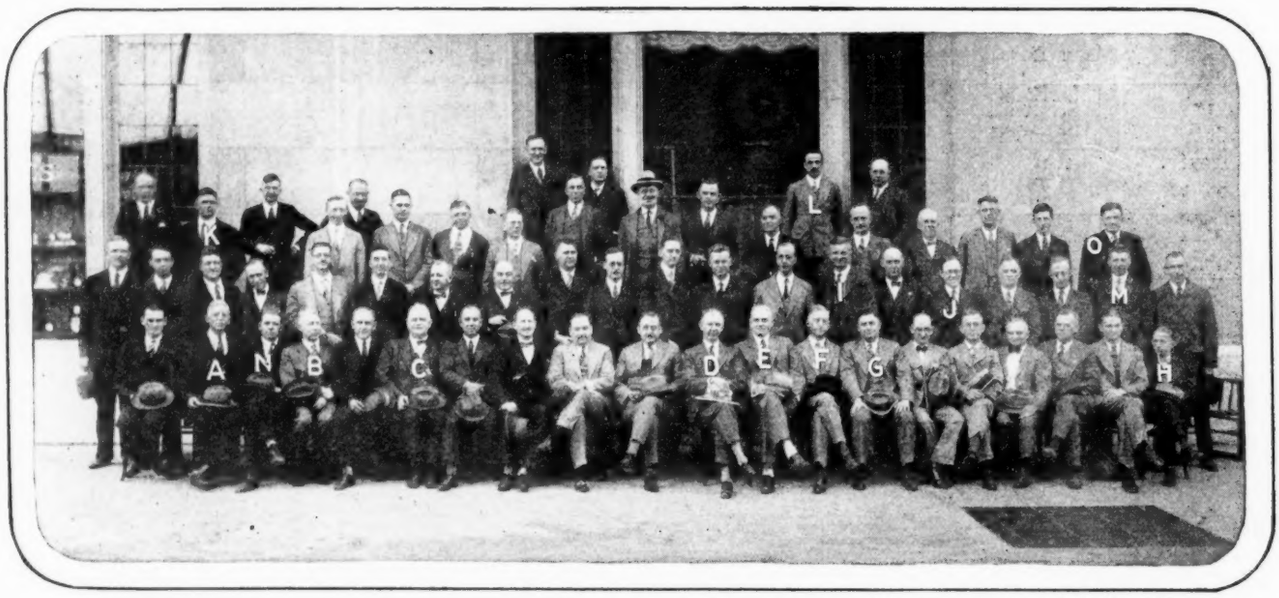
Fig 2—A Diesel in Key West, Fla., uses Bunker C oil of 12 deg. Beaumé gravity

even though no insurance was collected. It is inconceivable how a shutdown of over one hour could occur unless something serious had happened and in case of expensive breakage the insurance would go into force as indicated. Good business would seem to favor the elimination of stand-by units in many instances.

The elimination of stand-by units and the use of cheap fuel offers opportunities of large savings, but these are possible only through the employment of competent help. It is advisable, then, to be generous in the labor item, and in the plant under discussion, instead of a chief engineer whose services are worth but $350 per month, a $5,000 per year man should be engaged. Likewise, the day and night operating engineers should be $3,000 a year men. With this character of men the payroll becomes $1,170 per month, an increase of some $4,000 per year, but this makes possible a fuel saving of over $30,000.

All the opportunities for reduction in operating costs depend upon the intelligent attendance of the operators. Emphasis should be placed upon the desirability of abandoning the usual practice of employing anyone who happens around. The force should be as competent as can be procured, and a few dollars per week should not be withheld from a real capable man.

Seldom can the management hope to obtain low operating costs unless close track is kept of the power-house expenditures.

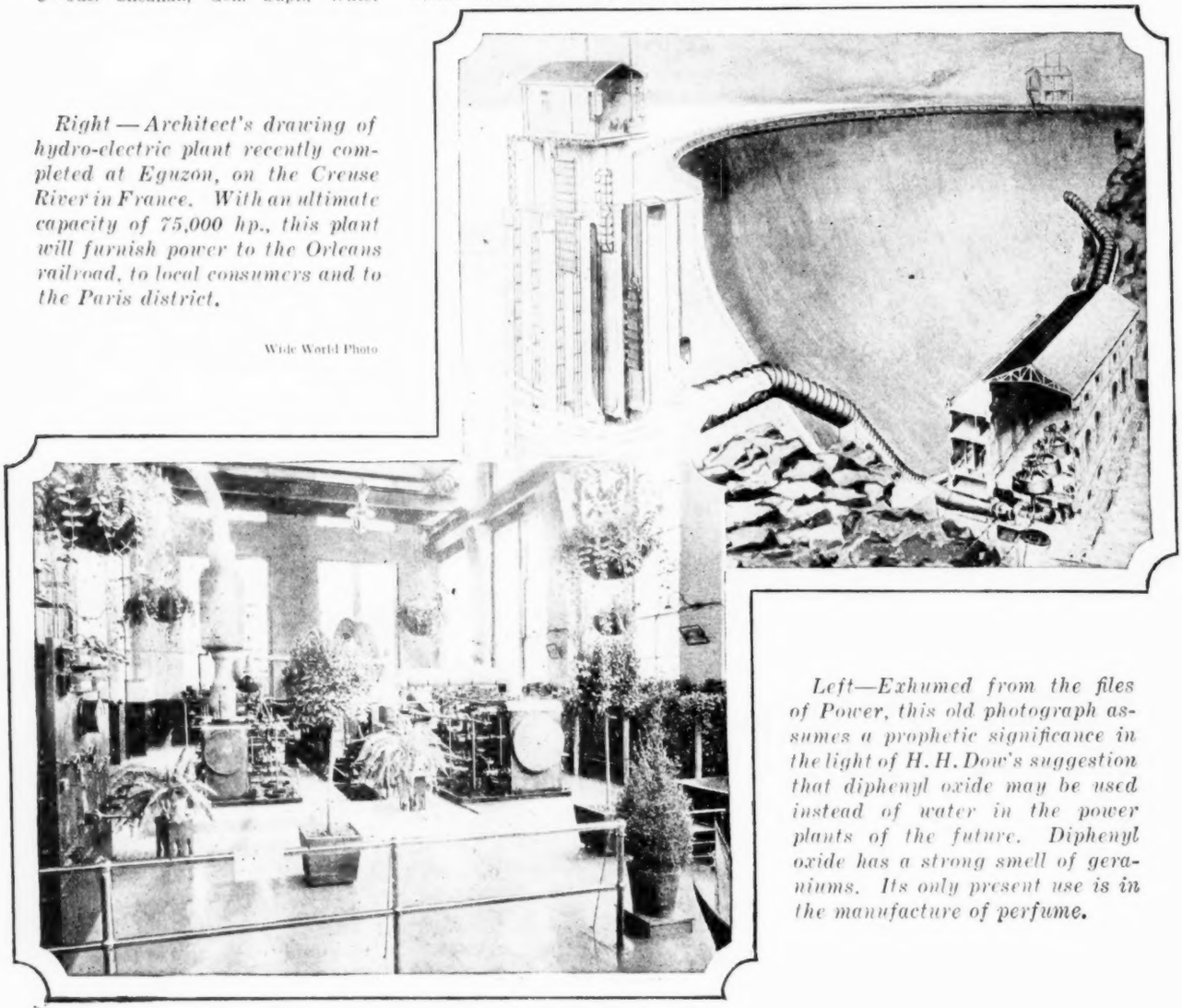


While others talk about putting engineering into government, Memphis (Tennessee) is doing it.

At the request of *Power* a group of members of the Memphis Engineers Club posed. The following engineers connected with the city government or departments appear in the picture: *B*—C. E. Davis, Chief Eng., Water Dept., *C*—Jas. Sheahan, Gen. Supt., Water Dept.; *D*—Major Thos. H. Allen, Commissioner of Fire and Police, and Vice Mayor; *E*—Rowlett Paine, Mayor; *F*—Major D. H. Connolly, President, Engineers Club; *G*—Wm. B. Fowler, Engr., Memphis Engineering Dept.; *H*—J. H. Ryan, Chief Draftsman, Eng. Dept.; *I*—W. G. Stromquist, Director Division of Sanitary Eng.; *J*—Harry N. Howe, Commissioner of Streets, Bridges and Sewers; *K*—D. C. Miller, Chief Clerk, Dept. Streets, Bridges and Sewers; *L*—J. J. Ryan, Asst. Supt. of Mains, Water Dept.; *O*—K. Markwell, Chief Draft., Eng. Dept.

Right — Architect's drawing of hydro-electric plant recently completed at Eguzon, on the Creuse River in France. With an ultimate capacity of 75,000 hp., this plant will furnish power to the Orleans railroad, to local consumers and to the Paris district.

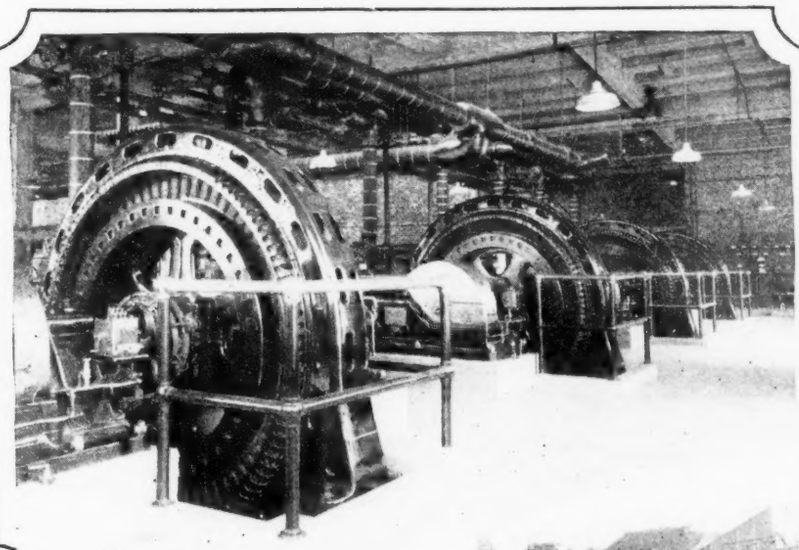
Wide World Photo



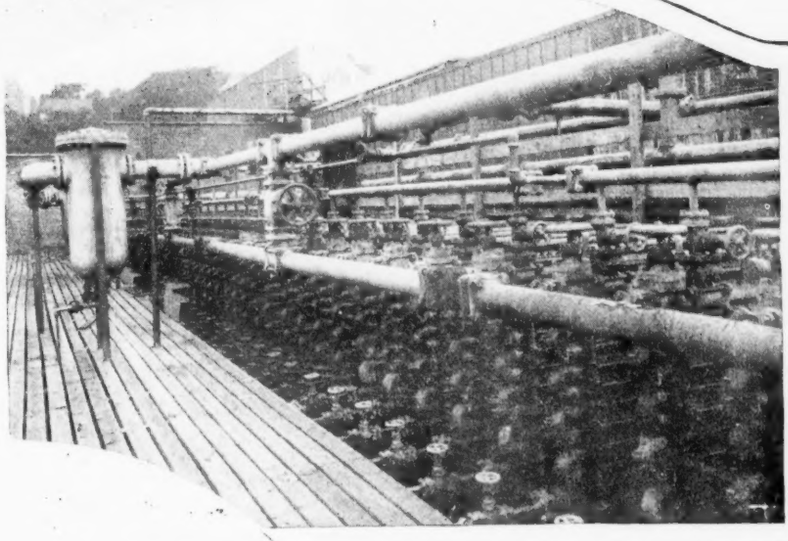
Left—Exhumed from the files of Power, this old photograph assumes a prophetic significance in the light of H. H. Dow's suggestion that diphenyl oxide may be used instead of water in the power plants of the future. Diphenyl oxide has a strong smell of geraniums. Its only present use is in the manufacture of perfume.

A Snappy Looking Ice Plant

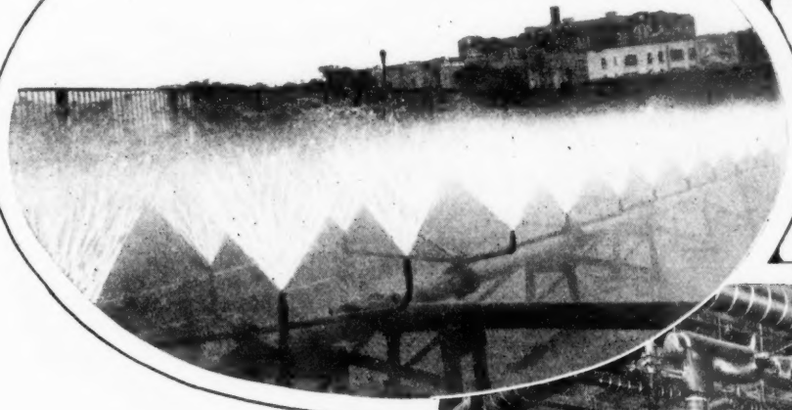
One must travel far to find a neater, cleaner, or better arranged plant than that of the Interborough Ice Co. at East 170th St. and Cromwell Ave., New York City.



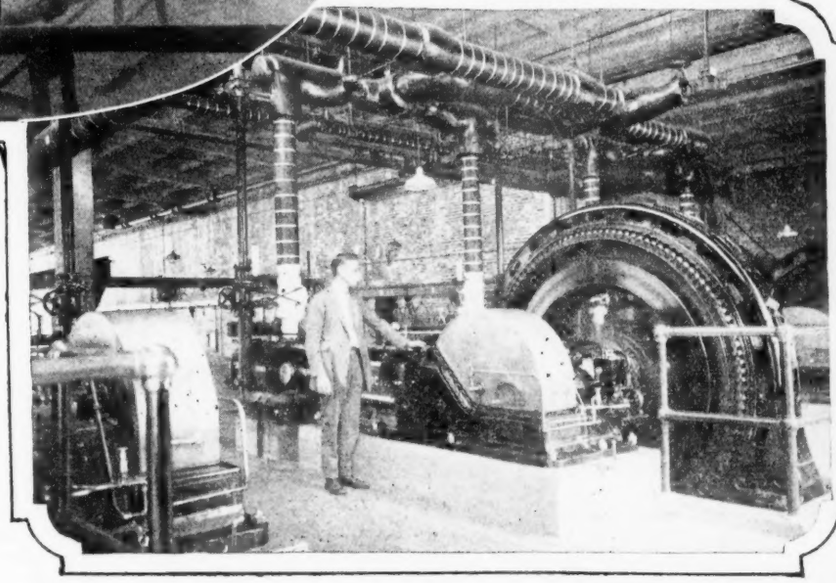
Left—Four synchronous-motor-driven twin-cylinder 50-ton ammonia compressors.



Right — Fifty-stand atmospheric type ammonia condenser at the Interborough Ice Plant. Four stands are used to precool the liquid ammonia.



Left—Spray pond on roof with 48 2-in. nozzles supplied by three motor-driven spray pumps with capacities of 2,000, 1,000 and 500 gal. per min.



Right—Charles Weder, Chief Engineer, standing beside one of his ice machines. Note the insulation job overhead.

*Boiler Feed-Water Purification—IX**

Priming and Foaming

Moisture in Steam Means Dirt and Dirt Spells Damage to Equipment. This Article Tells How to Attack Priming and Foaming and How to Get Rid of Those Impurities That Will Get by in Spite of Everything

By SHEPPARD T. POWELL
Consulting Chemical Engineer, Baltimore, Maryland

PRIMING and foaming are not synonymous terms, but are used loosely by some engineers to describe certain conditions of operation in which either one or the other of these troubles occur. Priming is the discharge of water from a boiler with the steam, by the more or less violent bursting of steam bubbles. It is similar to the "bumping" of water, noted frequently in an open receptacle. The phenomenon is accompanied by relatively great changes in the level of the surface of the water with such violent discharges of bursting bubbles as to carry water high into the steam space of the boiler, thereby causing "slugs" of water to be discharged from the boiler with the steam.

Foaming is the production of froth on the surface of the water. This condition is less violent than priming. The foam formed may stand only slightly above the surface of the water or it may fill the entire steam space.

Under extreme conditions priming and foaming may both occur at the same time. Water may be and usually is carried from boilers when there is no apparent indication of priming or foaming as such. The phenomenon is the result of the bursting of small bubbles of water at the surface, as the liquid is converted into vapor. The water so atomized has no opportunity to return to the body of the liquid and is carried along with the steam. These phenomena depend upon the construction and the operation of the boiler and the amount and character of the constituents in the boiling water.

The construction of the boiler has a marked effect on the priming and foaming tendencies of boiler water. Insufficient steam-disengaging space is one of the most important factors in this respect. Priming is greatly influenced by the rate of evaporation or sudden changes of the water level. Rapid heating of water, which results when a piece of scale is removed, may cause violent evaporation at one spot which will result in temporary priming. Cutting in a boiler quickly, causing a rapid drop in steam pressure, will also produce temporary priming of the boiler that is so operated. The size of the steam header and the velocity of the steam leaving the boiler and other factors accelerate both priming and foaming. The more obstruction within the boiler—staybolts, braces, baffles, etc.—the greater will be the tendency for these operating difficulties to occur.

The concentration of sodium salts in boiler water is held responsible for foaming and priming by a great many persons. There is little doubt that where such difficulties have been experienced, high concentrations of these salts have occurred. There is ample reliable data to prove, however, that it is possible to have excessively large amounts of sodium salts present in boiler water without these troubles occurring. Recently, experience and special studies have confirmed the theory that priming and foaming are due, not only to the concentration of sodium or other soluble compounds, but to the presence of these substances in relatively high concentrations in addition to suspended matter.

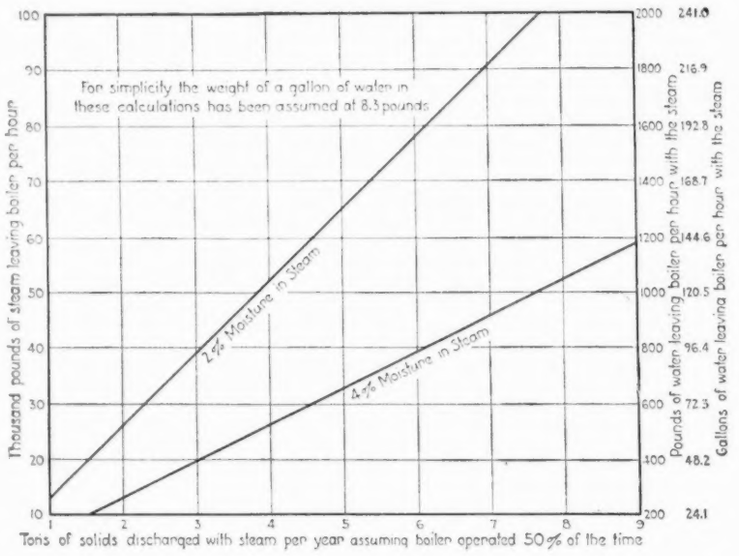

For simplicity the weight of a gallon of water in these calculations has been assumed at 8.3 pounds
2% Moisture in Steam
4% Moisture in Steam
Thousand pounds of steam leaving boiler per hour
Pounds of water leaving boiler per hour with the steam
Gallons of water leaving boiler per hour with the steam
Tons of solids discharged with steam per year assuming boiler operated 50% of the time

Fig. 1—Approximate weight of solids discharged from boiler with wet steam when boiler water contains 100 grains of solids per gallon

The fundamental physico-chemical theory to account for the formation of foam on water under steaming conditions in boiler operation has not been completely established. While there is still much to be determined concerning these phenomena, it is reasonably certain that the basic principle of foam formation is related closely to the surface tension of the liquid. Stated briefly, the foaming of boiler water may be expected where the surface tension of the liquid is changed, that is, either increased or decreased. In addition to this

*Previous installments have covered the following subjects: July 6, "Natural Waters and Their Impurities"; July 13, "Getting Rid of Impurities by Sedimentation and Coagulation"; July 20, "Filtration by Gravity and Pressure Filters"; July 27, "Softening Water by Chemicals"; Aug. 3, "Hot-Process Continuous Softeners"; Aug. 10, "Zeolites Explained"; Aug. 17, "Where Zeolites Fit in"; Aug. 24, "Boiler Compounds."

condition it is necessary, in order to support foam, that the films on the bubbles be sufficiently strong or stable to remain for a time before they burst. If these conditions are not fulfilled, the water will produce no foam, since the bubbles burst as soon as they are formed. Finely divided suspended matter stabilizes the film surfaces of the bubbles and to this extent accelerates foaming of the water.

All boiler water, even when the makeup is from evaporators, will contain some suspended solids. The nature and amounts of the solids vary greatly. Mud, sewage wastes, certain types of boiler compound, loose scale, calcium and magnesium compounds and many other substances, when present as suspended matter, will cause foaming under some conditions.

A number of plans have been advanced to inhibit the entrainment of water and solids with the steam discharged from boilers. They may be all classified under the following methods: (a) Controlling the concentration of suspended and soluble solids in the boiler water, by blowing down; (b) controlling the rate of steam production; (c) the mechanical purification of the steam; (d) the use of anti-foaming compounds.

Where foaming is experienced, treatment of the water should receive careful consideration. It is difficult to establish definite limits to the amount of soluble or suspended solids for the prevention of foaming. The character of the suspended matter is an important factor in this respect. As a general condition, other things being equal, the finer the suspended particles, the greater will be the foaming tendency of the boiler water. Solids may be present in the water due to surface washing of the watershed or to trade waste products. Frequently, suspended matter may become highly concentrated in the boiler owing to the precipitation of scale-forming solids or to the disintegration of old scale that has been loosened from tubes or drums and has not been blown from the boiler. A case of serious foaming was encountered in a small boiler plant a short time ago by the author. It was later determined that the difficulty was caused by the removal of large amounts of old scale, as a result of using zeolite-softened water in place of a water supply that contained an appreciable amount of scale-forming solids. The foaming disappeared as soon as the old scale was entirely removed from the tubes and drum of the installation. At times, foaming also is the result of poor operation of softening systems or is due to the use of certain compounds.

The most efficient means of preventing foaming is to reduce suspended solids to the lowest economical point by external treatment. The same applies to the soluble solids, which may be precipitated within the boiler. The concentration of the soluble salts in the boiler may be controlled by proper and intelligent blowing down, based on the analyses of the boiler water.

Much can be done to prevent foaming and especially priming of a boiler by the way in which it is operated. These troubles occur more frequently during sudden opening of the valves and while increasing rating rapidly than at other times. Care in operation in these respects will greatly lessen these difficulties. Carrying water in drums too high and forcing one side of a boiler is often responsible for this trouble, where the condition of the water is such as to favor high moisture in the steam. Whenever moisture is carried from a boiler with steam, some solids will be present also. The amount of these constituents will depend, of course, upon their concentration in the water.

In some boilers the amount of solids removed from the boiler with wet steam amounts to thousands of pounds in a year, even though there is no marked evidence of priming or foaming. In Fig. 1 is plotted the approximate weight of solids which may be carried from a boiler in which the concentration has reached 100 grains per gallon. For descriptive purposes the

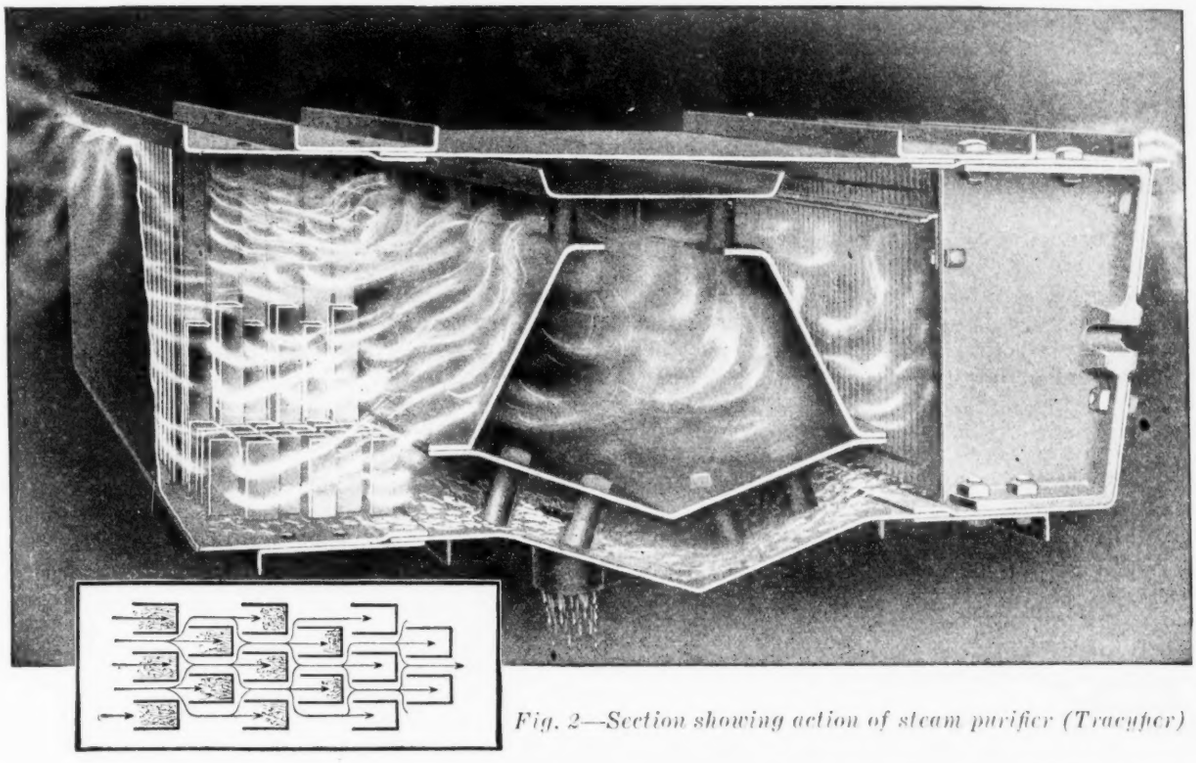


Fig. 2—Section showing action of steam purifier (Tracufier)

solids have been calculated for 2 per cent and for 4 per cent moisture in the steam. With a large boiler several tons of solids may be carried over in a year.

A number of appliances for the removal of moisture from the steam as it passes from the steam drum have been placed on the market. These appurtenances are of value generally and especially where it is difficult to prevent priming and foaming by treatment of the feed water or careful operation of the boilers. The removal of moisture from steam is effected by forcing the steam through appliances that bring about rapid changes of direction of flow. As a result of this sudden change in direction of flow the water in the steam impinges on the metal surface and flows back into the drum or is discharged outside the boiler.

How Steam Purifiers Operate

A section of a Tracy steam purifier, indicating the passage of steam through the apparatus, is shown in Fig. 2. The steam with its entrained moisture enters the purifier, as indicated by the arrows, and the direction of flow is changed several times before passing from the boiler. By this means the water is thrown out and runs back into the drum through the drain pipe in the bottom of the purifier. The apparatus consists of box like sections containing a number of vertical baffles shaped like gutters. The top and both ends of the separator are closed so that the steam is forced to enter the apparatus at each side, through the spaces between the vertical baffles. The steam flows thence into the central passageway connected with the bottom of the main steam outlet. Fig. 3 shows this purifier in place in a steam drum.

The Hagan steam purifier is shown in Figs. 4 and 5. The moisture is removed by the centrifugal force imparted to the steam as it is directed through the stationary turbine blades.

Referring to Fig. 5, the apparatus functions as follows: The moisture-laden steam passes into the separator through the turbine *B*. The resulting high centrifugal velocity of the steam causes the moisture to be thrown against the wall of the bowl of the separator. It then flows downward and returns to the boiler through the tangential drain *D*, while the steam turns and passes upward. The steam does not come again into contact with the water that has been thrown against the wall of the bowl.

A slight amount of water is re-entrained by the vortex motion at the bottom of the bowl. The steam leaving the separator at *E* throws this water out at the collecting ring *F*. The removal of water at this point is the result of the whirling motion of the steam as it travels upward.

There are certain waters, particularly those high in bicarbonate of soda or other sodium compounds, which may be used only by employing anti-foaming compounds.

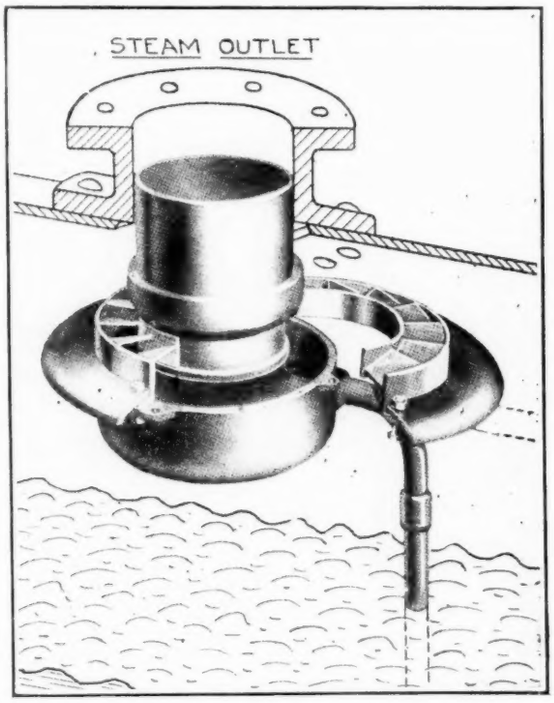


Fig. 4—Centrifugal steam purifier (Hagan) in steam drum of boiler

These products, which have been discussed previously,[1] are generally coagulants and emulsions of castor oil and organic substances. Preparations of this kind are used extensively in the treatment of feed water for railroads. In some sections of the country this form of treatment is essential, since no other treatment is applicable to correct the difficulties. Anti-foaming compounds, when properly used, may prevent foaming and priming. They will not eliminate wet steam completely, since this condition may and does occur in the absence of any evidence of boiler waters showing definite priming or foaming characteristics.

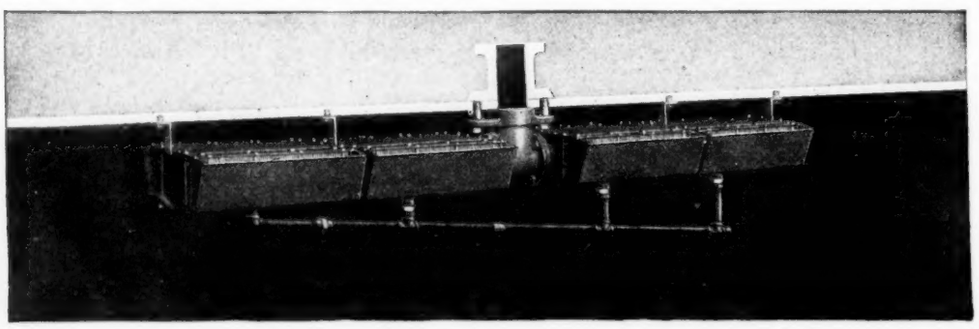


Fig. 3—Tracyfiers in place in drum of a boiler

The detrimental effects that result from the use of wet steam are great. Many serious accidents and numerous operating difficulties may be traced to this source. All water that leaves the boiler through the steam line carries with it solids either in suspension or in solution. It may be readily conceived that solids entering steam lines, turbines, engines, etc., create a potentially troublesome and dangerous situation.

[1] Aug. 24 issue.

Since the temperature of the steam is increased in passing through superheaters, any moisture that the steam contains will be evaporated, leaving a residue of solid matter. When this condition results, the solids are either deposited in the superheater tubes or carried through the superheater to be deposited at some other point along the path the steam travels. Whether or not the solids collect in the superheaters will depend primarily on the velocity of the steam and the construction of the apparatus. The nature of the solids has a considerable influence also upon the amount and character of the deposits in these tubes. They cause reduced heat-transfer efficiencies and may result in burned-out tubes. A further disadvantage of moisture in steam entering superheaters is the reduction in superheat caused by energy required to evaporate the water.

The serious results attending the presence of water in steam carried into reciprocating engines should require no special warning, since the history of steam engines abounds with cases of blown-out cylinder heads and similar accidents.

Wet steam, whether caused by definite priming or foaming action or merely the result of the mechanical removal of the boiler water due to high velocity of steam travel, is a phase of boiler-water control that deserves the most careful consideration of the steam-station operators. Aside from the serious accidents that may and do result as direct sequence to wet steam, many minor operating difficulties are encountered which over a year's time prove unduly costly.

The elimination of all moisture from steam as it leaves the boiler is essential to efficient operation. To accomplish this requires careful treatment of the feed-water makeup and a conscientious and an intelligent schedule of blowing down of the boilers. In addition to these precautions steam driers are usually desirable and frequently necessary, because steam as produced almost invariably contains some moisture even under the best conditions.

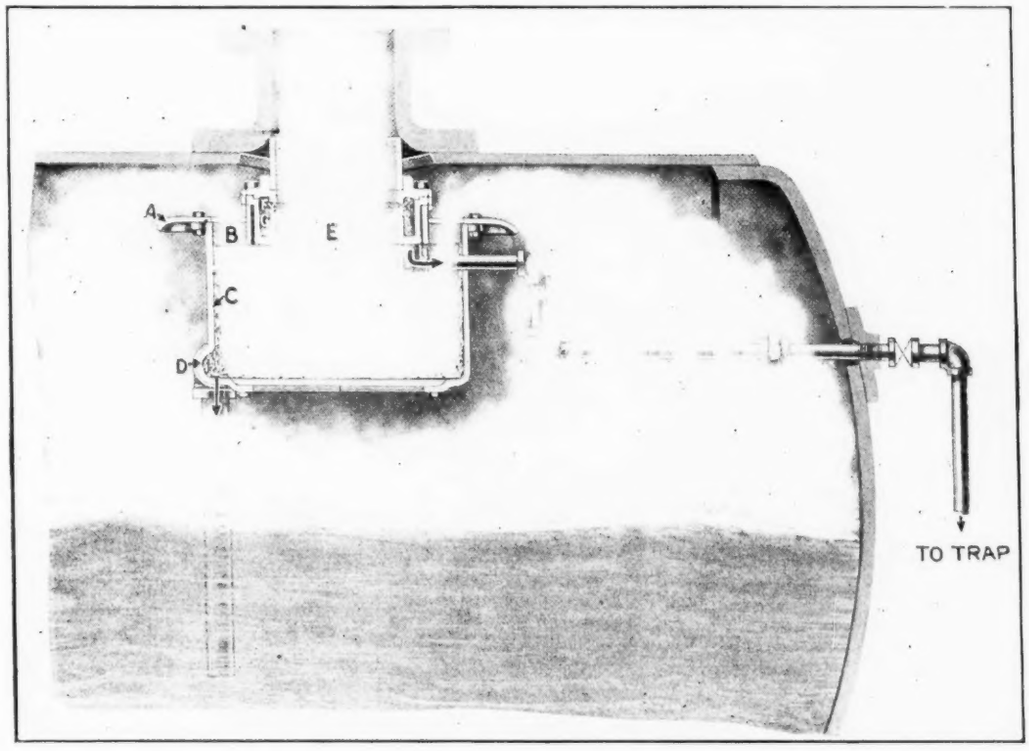


Fig. 5—Detail of Hagan purifier, showing passage of steam through the apparatus

Wet steam is excessively costly in the operation of turbines, owing to the eroding of turbine blades. This condition is generally due not only to the effect of the water, but also to the abrasive action of the solids carried in it. Where water leaves the boiler in slugs, as occurs when definite priming action takes place, turbine blades may be torn away, resulting in much damage to the equipment.

Even though turbine blades are not torn away by the action of the water and solids in the steam, the gradual erosion of the blades results in reduced efficiency of the equipment. This item alone may be considerable over the life of the turbine, even though no actual failures have occurred.

One of the frequent operating difficulties from wet steam is the deposition of solids in valves of one kind or another about the plant. There are many accidents on record where non-return valves have failed to function at the desired time, owing to jamming as a result of deposited solids.

FOUNDATIONS UNDER PLASTIC-BUILT FURNACE WALLS must be rigid. Often the walls are built on poor foundations with the result that cracks develop after drying and sometimes only after firing. Examination will show whether these cracks are due to poor foundations or improper materials. The foundation should be level or, if pitched, it should slope away from the furnace interior. Corners of continuous walls should be reinforced with an additional thickness of plastic; usually about 2 in. will be sufficient for the customary walls of 9 to 18 in. thickness. To forget this is to invite cracks at the corners of the walls.—*Hugh E. Weightman.*

Little Things Often Lead to Disastrous Accidents

By WILLIAM H. BOEHM

THE AUTHOR, who is vice-president of The Fidelity and Casualty Co. of New York City, relates a number of incidents where carelessness in apparently unimportant details has resulted in boiler explosions and engine wrecks involving large losses.

A WORKMAN employed in the construction of a new factory building left a washer lying on one of the roof timbers. Two years later a heavy flywheel in this factory burst, wrecked the engine upon which it was mounted, and damaged the building to the extent of about $10,000. It had taken these two years for the washer to creep three inches to the edge of the roof timber. It fell between the gear wheels on the governor of the engine, put the governor out of commission and allowed the engine to race and burst its flywheel by centrifugal force. An inspector found the washer mashed by the teeth of the gear wheels, and in the dust on the roof timber he traced the path that it had traveled before it fell.

A small engine was operated in a room containing three large turbines; the small engine was not insured, but the turbines were insured under a flywheel policy. A strap on the connecting rod of this small engine broke and bent outward enough to strike the cast-iron case over the crank of the engine. A piece not much larger than one's hand was broken out of this case and sent sailing across the engine room. It struck the governor of one of the turbines and put it out of commission. This caused all the turbines to race, because they were electrically connected. The turbines were wrecked. The owner recovered for this loss, although he had no insurance on the small engine that caused it.

Broken Capscrews Cause Engine Wreck

One of the two capscrews in the setting-up wedge on the crankpin end of the connecting rod of a Corliss engine broke and so loosened the crankpin brasses that the piston struck the cylinder head. This forced the piston off its rod, and then steam pressure shot the piston clear through the cylinder and against a brick wall just back of the engine. No one was hurt. The engine was wrecked. The loss was recovered under a policy of engine insurance. Only two weeks later one of the two capscrews in the crosshead end of a connecting rod of a similar engine broke. It fell in the clearance space between the under side of the connecting rod and the upper side of the crosshead. The crosshead was broken and the engine was badly damaged.

A small bolt in the cam plate forming part of the cutoff motion on a Corliss engine broke and allowed the cam plate to swing into such a position that the engine took steam at full stroke at a time when the load was light. This allowed the engine to race and burst its flywheel. This engine was wrecked and another one was badly damaged by pieces hurled from the flywheel.

A nut on the threaded end of a safety-valve lifting rod was supposed to be securely held in place by a locknut. Vibration due to the seating of the safety valve freed the locknut. Both nuts were a loose fit and they gradually crawled down the thread until they got under the slot in the bonnet through which the safety-valve lifting lever passes. In that position they blocked the safety valve and prevented it from blowing off. At five o'clock one afternoon the engineer banked his fires and went to his home, some three blocks away. While eating supper, he was startled by a violent explosion. He ran to the plant and found the entire building had been so completely demolished that it was nothing but a mass of ruins. With all other outlets to the boiler closed and the safety valve blocked, just 28 minutes was required to accumulate enough pressure to explode this boiler.

After some repairs had been made to a steam boiler, it was subjected to a hydrostatic test. To apply this test, the safety valve was closed by a blank flange. The fireman thought that the flange had been removed when he began to fire the boiler, but a violent explosion occurred.

Welded Boiler Head Fails

A crack was discovered in the turn of the flange of the lower head in a large vertical boiler. An employee prided himself on his skill in autogenous welding and did what he considered to be a fine job in the welding of this crack. He was so well satisfied with it that, after discovering that similar cracks were beginning to form in the heads of three other boilers alongside of the first one, he welded them also. Two weeks later one of the boilers violently exploded, knocked down the other three and completely wrecked the building. One man was killed, it was not the man who did the welding. He has learned since that autogenous welding is prohibited by all state and boiler insurance inspectors for any part of a boiler subjected to pressure.

High-pressure steam was supplied through a reducing valve to some large low-pressure digesters operated in a packing plant. The reducing valve was temporarily out of order, and while undergoing minor repairs, the attendant throttled the high-pressure steam down to low pressure by the use of a globe valve on the pipe line. There was a safety valve on each digester and a small safety valve at the end of the low-pressure pipe line. Just before five o'clock the attendant left his place at the globe valve and seated himself at a desk, where he began to make out his report for the day. While he was so engaged, one of the digesters violently exploded and wrecked the plant. The globe

valve was nearly closed and the safety valves were not large enough to take care of the pressure that accumulated after all the apparatus that had been using steam was shut down. The attendant was hurled across the digester floor and showered with vaporized grease. But his injuries were so slight that he was able to take care of some other employees who were more badly injured.

In another case high-pressure steam was supplied through a reducing valve at low pressure to a paper machine containing thirty-three steam drying cylinders made of cast iron. The boilers supplying this steam, together with their steam pipes, were insured. The steam drying cylinders were not. A rupture of one of the insured pipes violently exploded twenty-two of the uninsured drying cylinders. The loss amounted to $50,000.

Engineers Should Take No Chances

It will be seen from the foregoing that minor accidents often lead to violent explosions and disastrous losses. Engineers in power plants should take no chances. If a bolt or capscrew breaks, it is not sufficient to put it back as it was before the accident occurred. The cause of the breakage should be determined. If due to weakness, a stronger bolt should be used. If the bolt is standard and a larger one cannot be used, then there may be substituted for it an automobile bolt. Such a bolt not only is stronger because the threads on it are not cut so deep, but being made of more ductile material, it is less likely to fracture by bending stresses. Broken capscrews in the setting-up wedges of connecting-rod ends may be replaced by through bolts. The bolts should be made of nickel steel, heat-treated and threaded to take a standard automobile castellated nut securely pinned on.

If a locknut works loose, it is only another demonstration that locknuts don't lock. Lock washers don't lock. That is, not for any appreciable length of time, because vibration soon loosens them. The best locknut is a castellated nut held in place by a cotter pin. Capscrews often can be locked by threading a steel wire through their heads. Sometimes a plate washer is employed, the practice being to secure the washer in place by a steel pin and then to upset the washer against the side of the nut or bolt head. A thoughtful engineer using a little ingenuity usually can devise some positive means of preventing accidents of this sort. In any event it is always best to talk over such matters with an experienced insurance inspector. The business of the inspector is the prevention of accidents. A good inspector has a clear vision of the future and is able to conceive the sort of a disaster that may happen through the neglect of what might appear at the time to be a trivial matter.

Commutator Surfaces that have been turned true at low speeds often run out of true to some extent at high speeds, owing to centrifugal forces of the revolving element. Grinding should be done at full speed, which will thus take care of all deflections and produce a surface that is true at normal speed.

If Care Is Taken to keep the connecting rod to its original length as near as possible when adjusting the brasses, the clearances will remain unaltered and no shoulder will be formed and the risk of piston ring breakage will be reduced to a minimum.

Boiler Arch Problems

By W. F. Schaphorst

The correct design and location of boiler arches involves a number of problems, some of which are not always easily solved. The most important are:

Conduction of heat through the arch itself should be eliminated. Dead air spaces should be provided if possible. As a result the temperature of the outside walls becomes almost that of the boiler room. The furnace blocks should be so arranged as to prevent fire grooves, which in turn tend to cause spawling and checking of the blocks. This may be accomplished by means of staggered joints or wavy interlocking surfaces. The blocks should be so constructed that in case of spawling they will not drop out of the arch.

Construction should be such that high and low temperatures in the boiler room and the furnace will not come in contact with the same block. Such a condition will cause unequal expansion and contraction in the same block.

The design should provide for easy and convenient repairs to confine the "out of service" period to the time between peak loads. A frequent cause of old-style arch failures is the erosive action at the hip near the skew-back which takes the form of an ever-increasing groove. As extreme furnace temperatures are dealt with which are in excess of the "Softening point" of ordinary refractories, the usual sagging under these conditions only hastens eventual collapse. When this occurs, it requires the rebuilding of the whole arch or back ring with the bonding of old tile or brick to new, which is difficult to do without subsequent cracking. Besides, the "keying-up" with the curtain-wall support arch in place is by no means an easy matter.

This grooving action is present also with flat arches, generally occurring in the middle of the arch where the maximum temperature stratum is encountered. The repairing or patching of this advanced condition is a simple matter involving only the renewal of the affected tile. With the flat arch the labor charge making this repair is far less than repairing a similar condition with old-style arches. An old-style arch often cannot be repaired without complete rebuilding.

How the Nature of Fuel To Be Burned Affects the Design

What kind of fuel will be burned? This involves consideration as to whether the percentage of volatile matter is high or low; moisture; ash; size of fuel, etc. The general conditions referred to in this article are those obtained with bituminous or more specifically Mid-Western fuels. The conditions prevailing with the use of coke breeze, while somewhat different, do not seriously affect the furnace design. In fact, this fuel is in many instances used as an alternate on forced-draft installations. The burning of anthracite fuel on blast-furnace stokers presents certain special problems that cannot be dealt with in detail here. Even with similar classes of fuels there exist special characteristics of certain coals from the same district, and the successful burning of them may mean alterations in furnace design. It is obviously not easy to formulate a code of furnace practice to cover all special conditions, but on the other hand those principles established by usage and corroborated by experiment should offer the furnace designer considerable insurance against failure.

Simple Directions for

Aligning a Horizontal Engine

By WM. BREHMER

OCCASIONALLY, an engine in service will give much trouble due to poor alignment of the working parts. Perhaps some error was made in the original lining up and erection. For example, the engine bed may not have been properly supported and so it became distorted when the grouting was poured; or the anchor bolts may have been pulled up before the cement had hardened. Excessive wear has resulted and superficial attempts to remedy this condition have, of course, failed to remove the cause. In any case, if an engine gives persistent trouble that defies ordinary methods of remedy, the operator will do well to make a thorough examination of the alignment.

Although an approximate alignment test can be made without the removal of any of the running parts more accurate results can be obtained if the shaft, connecting-rod, crosshead, piston, stuffing box and cylinder head are removed. Incidentally, this dismantling may afford an opportunity for a much-needed examination of all wearing surfaces for signs of excessive friction.

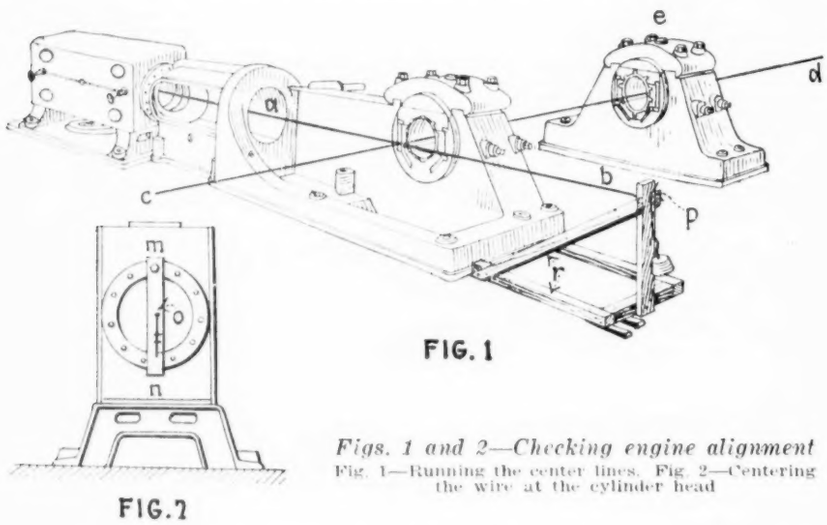


FIG. 1

FIG. 2

Figs. 1 and 2—Checking engine alignment
Fig. 1—Running the center lines. Fig. 2—Centering the wire at the cylinder head

The general method of alignment is indicated in Fig. 1. The first step is to run a fine steel wire *ab* and to center it accurately along the axis of the engine cylinder. A similar test wire *cd* is run through the centers of the main bearings. These two reference lines will enable one to test all the main parts of the engine.

To run the steel wire *ab*, first secure a 1½x1½-in. or a 2x2-in. stick *mn* and bolt this by one bolt through an oversize hole to the head-end cylinder flange, as in Fig. 2. Locate point *O* on this stick at the center of the cylinder bore. Drill a small hole through the board at this point, insert the wire through this hole and nail the end of the wire to the board. By tapping the stick with a hammer, one can accurately center this end of the wire to within a few thousandths of an inch of the center of the cylinder bore. It is desirable to fasten the other end of the wire *b* close to the engine. This is because the wire will tend to sag, and if it is quite long this sagging may introduce an appreciable error. A good way is to erect a simple though firm wooden stand, as shown in Fig. 1. Measure the height of point *O* on the stick *mn* above the floor. And at the same height above the floor drill a hole *P* in the stand. Run the wire through the hole and hang a heavy weight on the end of it.

Now, by means of inside calipers, test the position of the wire in the cylinder at both ends. First, center the wire at the head end by tapping the stick *mn* into position. The centering must be done with great care. Then center it at the crank end by adjusting the position of the stand. This may be raised off the floor but not lowered. Hence it might be necessary to drill another hole for the wire. By drilling somewhat too low, it is easy enough to raise the stand to the right height by wedges. It is to be noted that the angle *r* of the sloping braces should be less than 45 deg. to prevent tipping of the stand.

While the wire is being centered in the cylinder, it might be found that the cylinder has been worn considerably out of round. Obviously, the weight of the piston would tend to cause wear on the bottom cylinder surface. Again, if the crosshead guides have been out of line, this will have caused the piston rings to scrape the cylinder wall. Hence, the careful testing of the cylinder wall may give a valuable clue to poor alignment of the crosshead guides or to a kink in the piston rod.

Having established the line *ab* through the engine cylinder, the next step is to check up the alignment of the crankshaft bearings. This test should be made before that of the crosshead guides. In a properly aligned engine the crankshaft should be at right angles to the cylinder axis and the cylinder and shaft axes should lie in the same horizontal plane. That is, the lines *ab* and *cd* should touch each other, be level, and form a 90-deg. angle.

The question whether the lines *ab* and *cd* are at right angles may be decided by means of a large square. To make an accurate measurement, the square should be supported in some way and not simply held in the air by hand.

The angle formed by the two wires can best be checked by the 3–4–5 method. From the intersection of the two wires lay out along *ab* a distance of 4 ft. and on *cd* a distance of 3 ft. Measure the diagonal between

the two points thus determined, and if the wires are at 90-deg. angles, the diagonals will measure 5 ft. This method is illustrated in Fig. 3.

To test the alignment of the crosshead guides, with a pair of calipers or inside micrometers measure the distances from the top and bottom guides to the wire *ab*. If the distances at the front and back vary, the frame is not square with the cylinder, due either to guide wear or to misalignment of the cylinder flange.

If the measurements have been carefully made and checked against each other, one may proceed with the readjustment of the main wearing surfaces. The

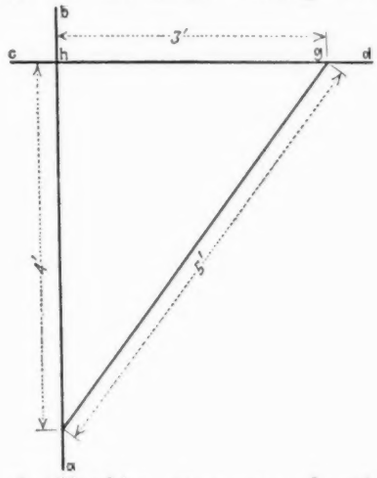


Fig. 3—Checking squareness of center lines

chances are that only one of these is out of line. Good judgment must be used in deciding which surface offers the best chance of truing up the engine. In any case only one adjustment at a time should be attempted.

Reducing Costs With Old Equipment

By A. F. Sheehan

In industrial plants there exist countless chances for substantial reduction in power costs that can be seen by the process of considering every possible angle to the power problem. It is too often taken for granted that a plant over twenty years old must be replaced by more efficient equipment, and the high initial cost of this replacement often results in a decision to abandon the generation of power as far as possible. As all know, it costs money to change equipment, but a thing that sometimes saves nearly as much and costs nothing to put into practice, is the changing of methods of operating the equipment. This idea is so applicable to many plants that it is surprising that it is so little followed.

A paper mill operating three paper machines driven by slide-valve engines whose combined power output was 255 hp., was purchasing the power for other departments at a rate of 1.2c. per kw.-hr. They had in the plant a 750-hp. simple Corliss that had been shut down on account of the high steam rate, and as it was too small to handle the total load, either a connection with the power company or a new engine was necessary. No capital being available for power equipment and a favorable rate being offered, they signed a contract for a year. The fuel saving about balanced the electric bills, and the labor saving in the boiler room was figured to be the net saving, around $3,000 a year. In addition to this they could shut down the oldest boilers, keeping only the most efficient in service. The steam flow to the machine room was 13,000 lb. per hour, used mostly by the engines that were retained because all the exhaust could be used on drying paper. The boiler could be operated at 165 lb. pressure, but after shutting down the main engine the pressure was reduced to 100 lb.

The engineer reported that the main engine was in good condition, but the steam rate had been high on account of overload. He showed indicator diagrams from the paper-machine engines of the throttling-governor type, and the load was such that not over 45-lb. steam pressure existed in the cylinders at any time. By operating the boilers at maximum pressure, the 13,000 lb. of steam used in the drier could first be expanded from 165 lb. down to 45 lb. in the main engine and exhausted to the smaller engines without interfering with their power output, as the load was steady in the machine room. Under these conditions the main engine could develop 300 hp. at a very slight increase in fuel consumption, and as operations were continuous for 24 hours daily and 300 days a year, the reduction in power cost would be well worth while. As about 200 kw. would be generated, the saving would be 1,440,000 kw.-hr. Allowing for an increase of 7 per cent in steam flow due to the heat extracted from the steam in the main engine, the power bill would be reduced $17,000 a year by using only $3,000 more coal. Making all possible allowances, a net saving of $10,000 a year was possible without any initial outlay and no increase in labor costs, as the same crew is required in either case. Thus an inefficient generating unit could be converted into a very efficient one, but with reduced capacity.

A textile plant had a 1,000-hp. four-valve engine with two high-pressure cylinders; one boiler having been condemned, necessitated purchased power to some extent owing to lack of boiler capacity. As the power company's line crossed a near-by property, they agreed to bear all expenses of the electrical installation except the motor cost and loaned the money for the latter purpose. As one cylinder had operated condensing and the other at 10 lb. back pressure for boiling kiers and drying cloth, the condensing cylinder was disconnected and the load on the other reduced to meet the requirements for exhaust steam.

Being a resourceful engineer, he planned further and hit upon a way to secure maximum power from the process steam. The cylinder operating at 10 lb. back pressure developed about 300 hp. He decided to expand all process steam in one cylinder to 50 lb. back pressure, also increasing engine speed 20 per cent, the remaining cylinder to take half the exhaust from the high-pressure cylinder and the other half to be extracted between cylinders and used in the process that demanded that pressure. He found that passing enough steam for all processes through the first cylinder at maximum boiler pressure gave an economical point of cutoff and a terminal pressure that insured minimum pressure drop between the cylinders.

The power rates in this plant are not known and the exact saving cannot be given, but it is evidently large, as the power output from process steam jumped from 300 to 500 hp. Cutting 200 hp. from the power bill, 9 to 12 hours daily, is surely a creditable showing for an out-of-date plant.

Joint Efficiency Easily Checked With New Tables

By A. G. PETERSE
Boiler Inspector, The Ocean Accident & Guarantee Corp., Ltd.

TO DETERMINE the safe working pressure of a boiler, it is necessary to know the efficiency of its riveted joints, especially of the longitudinal seams. The highest efficiency is obtained when the strength of the plate between the rivet holes is equal to the shearing strength of the rivets, the crushing strength of the plate in front of the rivets or such a combination.

It appears that the weakest part of the greater number of joints is the strength of the plate between the rivet holes. When this is the case, the efficiency may easily be computed by using the short formula, namely:

$$\frac{Pitch - diameter\ rivet\ hole}{Pitch} = \frac{P - D}{P} = Efficiency$$

However, when a joint does not fail in this way different formulas must be used. These involve a number of factors that are cumbersome to handle. In reviewing various calculations, it is noted that when a joint will fail by the shearing of the rivets, crushing of the plate or a combination of these two causes, the numerators of the formulas remain the same, only the denominators change with the pitch. This fact makes it possible to substitute constants for the numerators in such calculations.

Tables I and II were compiled to eliminate the tedious calculations necessary to determine definitely the weakest part of the joint and the efficiency. The tables cover the types of joints and the various combinations of plate thicknesses and rivet holes suggested in the A.S.M.E. Boiler Construction Code.

Table I gives the maximum efficiency, the critical pitch at which the maximum efficiency occurs and the "constant." If the pitch of a joint is greater than the one shown in Table I, the constant for that joint should be divided by the pitch to obtain the efficiency.

$$\frac{Constant}{Pitch} = Efficiency$$

Table II contains the efficiencies of all joints where the pitch is smaller than the critical pitch as shown in Table I. In other words, the efficiency of joints that can be calculated by using the short formula $\frac{P - D}{P}$, can be taken from Table II by merely reading the efficiency corresponding to the diameter of the rivet hole and the pitch. Should the pitch be greater than the critical pitch for a joint, Table II cannot be used, as it would give a too-high efficiency, but the constant in Table I must be used, as explained before.

TABLE I—MAXIMUM EFFICIENCIES AND CONSTANTS FOR BOILER JOINTS

The efficiency of a riveted joint, having a pitch smaller than given in this table = $\frac{\text{Pitch} - \text{Rivet hole}}{\text{Pitch}}$

The efficiency in percentage of a riveted joint, having a pitch greater than given in this table = $\frac{\text{Constant}}{\text{Pitch}}$

Plate Thickness	Diameter Rivet Hole	Lap Joints		Butt Joints with Double Straps of Unequal Width			
		Single	Double	Double	Triple	Quadruple	
0.25	0.688	1.87	3.07	4.25	6.62	13.75	Pitch
		63.3%	77.6%	83.8%	89.6%	95%	Maximum Efficiency
		118.7	238	356	593	1,306	Constant
0.281	0.688	1.74	2.81	4.12	6.50	13.36	Pitch
		60.6%	75.5%	83.3%	89.4%	94.8%	Maximum Efficiency
		105.6	212	343	581	1,267	Constant
0.312	0.75	1.88	3.01	4.47	7.06	14.50	Pitch
		60.2%	75.1%	83.3%	89.4%	94.8%	Maximum Efficiency
		113.3	226	372	631	1,375	Constant
0.344	0.75	1.78	2.80	4.37	6.96	14.17	Pitch
		57.8%	73.3%	82.8%	89.2%	94.7%	Maximum Efficiency
		102.9	205	362	621	1,342	Constant
0.375	0.812	1.92	3.02	4.72	7.54	15.51	Pitch
		57.7%	73.1%	82.8%	89.3%	94.7%	Maximum Efficiency
		110.6	221	391	673	1,470	Constant
0.406	0.812	1.84	2.85	4.64	7.45	15.26	Pitch
		55.8%	71.5%	82.5%	89.1%	94.7%	Maximum Efficiency
		102.4	204	383	664	1,445	Constant
0.438	0.875	1.97	3.06	5	8.02	16.25	Pitch
		55.7%	71.5%	82.5%	89.1%	94.6%	Maximum Efficiency
		110	219	412.5	715	1,537	Constant
0.438	0.938	2.20	3.47	5.44	8.68	17.69	Pitch
		57.4%	73%	82.7%	89.2%	94.7%	Maximum Efficiency
		126.2	253	450	774	1,675	Constant
0.5	0.938	2.04	3.15	5.28	8.52	17.21	Pitch
		54.1%	70.2%	82.3%	89%	94.5%	Maximum Efficiency
		110.6	221	434	758	1,627	Contsant
0.562	1	2.11	3.24	5.57	9.02	18.17	Pitch
		52.7%	69.1%	82.1%	89%	94.5%	Maximum Efficiency
		111.7	224	457	802	1,717	Constant
0.562	1.062	2.33	3.58	6.01	9.67	19.54	Pitch
		54.2%	70.4%	82.3%	89%	94.6%	Maximum Efficiency
		126.2	252	495	861	1,848	Constant
0.625	1.062	2.20	3.33	5.88	9.54	19.13	Pitch
		51.6%	68.1%	81.9%	88.9%	94.4%	Maximum Efficiency
		113.7	227	482	848	1,807	Constant

Table does not apply for butt straps less than 0.364 of rivet hole. Calculations are based on A.S.M.E. Code requirements for steel riveted joints. Tensile strength 55,000 lb. per sq.in.; shearing strength, 44,000 lb.

TABLE II—EFFICIENCIES OF JOINTS, FAILING BETWEEN OUTER RIVET HOLES

Pitch, Inches	Diameter Rivet Holes: ⅝″ 0.625″	11/16″ 0.688″	¾″ 0.75″	13/16″ 0.812″	⅞″ 0.875″	15/16″ 0.938	1″ 1″	1 1/16″ 1.062″
1½	58.3 (1/4)	54.2	50	45.8	41.6	37.5	33.3	29.2
⅝	61.5	57.7 (9/32)	53.8	50	46.1	42.3	38.5	34.6
¾	64.3	60.7	57.1 (11/32)	53.5 (13/32)	50	46.4	42.8	39.3
⅞	66.6	63.3 (1/4)	60 (5/16)	56.6 (3/8)	53.3 (7/16)	50	46.6	43.3
2	68.7	65.6	62.5	59.4	56.2	53.1 (1/2)	50 (9/16)	46.9
⅛	70.6	67.6	64.7	61.7	58.8	55.8 (7/16)	53	50 (1/2)
¼	72.2	69.4	66.6	63.8	61.1	58.3	55.5	52.8 (9/16)
⅜	73.6 (9/32)	71	68.4	63.7	63.1	60.5	57.8	55.2
½	75 (1/4)	72.5	70	67.5	65	62.4	60	57.5
⅝	76.2	73.8	71.4	69	66.6	64.2	61.9	59.5
¾	77.2	75 (9/32)	72.7 (11/32)	70.4 (13/32)	68.1	65.9	63.6	61.3
⅞	78.2	76.1	73.9	71.7	69.5	67.4	65.2	63
3	79.1	77.1 (1/4)	75 (5/16)	72.9 (3/8)	70.8 (7/16)	68.7	66.6	64.5
⅛	80	78	76	74	72	70 (1/2)	68 (9/16)	66
¼	80.7	78.8	76.9	75	73	71.1	69.2	67.3 (5/8)
⅜	81.4	79.6	77.7	75.9	74	72.2 (7/16)	70.3	68.5
½	82.1	80.3	78.5	76.7	75	73.2	71.4	69.6 (9/16)
⅝	82.7 (9/32)	81	79.3	77.6	75.8	74.1	72.4	70.7
¾	83.3 (1/4)	81.6	80	78.3	76.6	75	73.3	71.6
⅞	83.8	82.2	80.6	79	77.4	75.8	74.2	72.5
4	84.3	82.8	81.2	79.6	78.1	76.5	75	73.4
⅛	84.8	83.3 (9/32)	81.8	80.3	78.8	77.2	75.7	74.2
¼	85.3	83.8 (1/4)	82.3	80.8	79.4	77.9	76.4	75
⅜	85.7	84.3	82.8 (11/32) (5/16)	81.4	80	78.5	77.1	75.7
½	86.1	84.7	83.3	81.9	80.5	79.1	77.7	76.4
⅝	86.5	85.1	83.7	82.4 (13/32) (3/8)	81	79.7	78.3	77
¾	86.8	85.5	84.2	82.8	81.5	80.2	78.9	77.6
⅞	87.1	85.9	84.6	83.3	82	80.7	79.5	78.2
5	87.5	86.2	85	83.7	82.5 (7/16)	81.2	80	78.7
⅛	87.8	86.6	85.3	84.1	82.9	81.7	80.5	79.2
¼	88.1	86.9	85.7	84.5	83.3	82.1 (1/2)	80.9	79.7
⅜	88.3	87.2	86	84.8	83.7	82.5 (7/16)	81.4	80.2
½	88.6	87.5	86.3	85.2	84.1	82.9	81.8 (9/16)	80.6
⅝	88.9	87.7	86.6	85.5	84.4	83.3	82.2	81.1
¾	89.1 (9/32)	88	86.9	85.8	84.7	83.7	82.6	81.5
⅞	89.3 (1/4)	88.3	87.2	86.1	85.1	84	82.9	81.9 (5/8)
6	89.5	88.5	87.5	86.4	85.4	84.3	83.3	82.3 (9/16)
¼	90	89	88	87	86	85	84	83
½	90.3	89.4 (9/32) (1/4)	88.4	87.5	86.5	85.5	84.6	83.6
¾	90.7	89.8	88.8 (11/32)	87.9	87	86.1	85.1	84.2
7	91	90.1	89.2 (5/16)	88.3	87.5	86.6	85.7	84.8
¼	91.3	90.5	89.6	88.7 (13/32)	87.9	87	86.2	85.3
½	91.6	90.8	90	89.1 (3/8)	88.3	87.5	86.6	85.8
¾	91.9	91.1	90.3	89 5	88.7	87.9	87.1	86.3
8	92.1	91.4	90.6	89.8	89 (7/16)	88.2	87.5	86.7
¼	92.4	91.6	90.9	90.1	89.4	88.6	87.8	87.1
½	92.6	91.9	91.1	90.4	89.7	88.9 (1/2) (7/16)	88.2	87.5
¾	92.8	92.1	91.4	90.7	90	89.2	88.5	87.8
9	93	92.3	91.6	90.9	90.2	89.5	88.8 (9/16)	88.2
¼	93.2	92.5	91.8	91.2	90.5	89.8	89.2	88.5
½	93.4	92.7	92.1	91.4	90.7	90.1	89.4	88.8 (5/8) (9/16)
¾	93.5	92.9	92.3	91.6	91	90.3	89.7	89.1
10	93.7	93.1	92.5	91.8	91.2	90.6	90	89.3
½	94	93.4	92.8	92.2	91.6	91	90.4	89.8
11	94.3	93.7	93.1	92.6	92	91.4	90.9	90.3
½	94.5 (9/32)	94	93.4	92.9	92.3	91.8	91.3	90.7
12	94.7 (1/4)	94 2	93.7	93.2	92.7	92.1	91.6	91.1
½	....	94.5	94	93.5	93	92.5	92	91.5
13	....	94.7 (9/32)	94.2	93.7	93.2	92.7	92.3	91.8
½	....	94.9 (1/4)	94.4	93.9	93.5	93	92.5	92.1
14	....	....	94.6 (11/32)	94.2	93.7	93.3	92.8	92.4
½	....	....	94.8 (5/16)	94.4	93.9	93.5	93.1	92.6
15	....	....	....	94.5 (13/32)	94.1	93.7	93.3	92.9
½	....	....	....	94.7 (3/8)	94.3	93.9	93.5	93.1
16	....	....	....	....	94.5 (7/16)	94.1	93.7	93.3
½	....	....	....	....	....	94.3	93.9	93.5
17	....	....	....	....	....	94.4 (1/2)	94.1	93.7
½	....	....	....	....	....	94.6 (7/16)	94.2	93.9
18	....	....	....	....	....	....	94.4 (9/16)	94.1
½	....	....	....	....	....	....	....	94.2
19	....	....	....	....	....	....	....	94.4 (5/8)
½	....	....	....	....	....	....	....	94.5 (9/16)

Single-riveted lap joint

Double-riveted lap joint

Double-riveted two-strap butt joint

Triple-riveted two-strap butt joint

Quadruple-riveted two-strap butt joint

Figures in parentheses are plate thicknesses for which the given efficiency is the maximum possible. Table not to be used for larger pitches. Note that these limiting values run in diagonal bands corresponding to type of joint indicated on arrows

To avoid using Table II for a pitch that is too great, fractional figures indicating the plate thickness are placed in line with the critical pitch for each type of joint mentioned in Table I. For instance, looking down the column for $\frac{13}{16}$-in. rivet holes, it will be noted that with a plate thickness of $\frac{3}{4}$ in. the critical pitch for:

Single-riveted lap joints is at $1\frac{7}{8}$ in.
Double-riveted lap joints is at 3 in.
Double-riveted butt joints is at $4\frac{3}{8}$ in.
Triple-riveted butt joints is at $7\frac{1}{2}$ in.
Quadruple-riveted butt joints is at $15\frac{1}{2}$ in.

The succession in which these various types of joints are eliminated is the same in each column. Thus, Table II readily shows whether or not it can be used for a given pitch.

With these indications it will be found that Table II can be used without consulting Table I for the critical pitch. For example:

Given a double-riveted butt strap, $\frac{13}{16}$-in. rivet holes, 5-in.x$2\frac{1}{2}$-in. pitch and $\frac{7}{16}$-in. plate thickness. In the column for $\frac{13}{16}$-in. rivet holes in Table II, the answer is in line with the 5-in. pitch and reads 81.2 per cent. It will be noted this is within the limit for double-riveted butt joints having a $\frac{7}{16}$-in. plate thickness, indicated at 82.5 per cent, and is therefore correct.

Should the given joint have a pitch of $5\frac{1}{2}$ in. the efficiency found in Table II would be 82.9 per cent. This appears in that part of the column from which the double-riveted butt joints having a $\frac{7}{16}$-in. plate thickness have been eliminated, and therefore does not apply for this pitch.

In Table I it will be found that the maximum efficiency for the joint under discussion is exactly 82.7 per cent at a pitch of 5.44 in. The efficiency for $5\frac{1}{2}$-in. pitch therefore would be

$$\frac{Constant}{Pitch} = \frac{450}{5.5} = 81.8 \text{ per cent}$$

When one is not familiar with the indications in Table II it is always safe to consult Table I first and see whether the constant or Table II should be used for a given joint.

Sulphur Does Not Hurt Exhaust Valves

By Charles H. Bushnell

Sulphur has long been the pet evil spirit of Diesel engineers as destruction of exhaust valves was supposed to be one of its specialties, although it is by no means a fact.

An elementary knowledge of chemistry informs us that any sulphur in the fuel oil will burn to SO_2, or sulphur dioxide. As there is always an excess of oxygen together with H_2O, or water, formed by combustion of the hydrogen of the fuel, we have all the essential elements of sulphurous acid (H_2SO_3) or sulphuric acid (H_2SO_4). The essential conditions, however, are lacking, as the H_2O must be in the form of moisture before any reaction can take place. During normal operation it passes out the exhaust in the form of superheated steam.

As this fact became recognized, the explanation was somewhat altered. Any engine must be stopped at times, and this was supposed to allow the H_2O to combine with the SO_2 in an attack on the valves. There are some serious objections to this hypothesis: (a) Few engines stop without rolling a few revolutions after the fuel is cut off and this will thoroughly scavenge out any SO_2 there might be; (b) as a rule high-sulphur fuel is too viscous for the pumps to handle without preheating, so the engine must be stopped on light fuel containing little or no sulphur; (c) the valves burn while an engine is in operation and there is no cooling.

With all this the fact remains that fuel containing a high percentage of sulphur is often destructive to exhaust valves. If sulphur is not the cause, what is?

If we examine an exhaust valve that has been in use for some time, we find the seating surface covered with hard bright scale that may be chipped off. The seating surface of the cage will be found to have a similar though thinner scale. This scale is simply iron oxide, or what is often called fire scale. If we examine valves that have just begun to leak, we shall find one or more narrow grooves across the valve. These are usually, though not always, straight across. In the beginning they may be irregular, but will soon become straight. Once a valve starts to leak, it will burn rapidly because of the blowtorch effect during combustion. If the valve and cage are of the same material, it will be found that a leak seldom if ever starts at the valve seat of the cage; however, continued operation with a burned valve will burn the cage.

From this, one may draw certain conclusions: (a) Leaks start from a small amount of scale flaking off; (b) the scale must be of some thickness before a leak will be started. This is proved by the fact that leaks do not ordinarily start on the surface of the cage where the scale is thinner than on the valve.

The thickness of the scale for a given length of service will depend upon the operating temperature of the valve, the material of the valve, and the rate at which the scale wears away. If this be true, the life of the valve may be increased by reducing its operating temperature. This may be done either by more effective cooling or by reducing the temperature of the exhaust gases. In addition the valve life can be lengthened by using some material that is less readily oxidized, such as a special alloy steel. Reducing the width of the seat will cause the scale to wear away more rapidly by means of the greater bearing pressure. This may easily be overdone, for if the seat is too narrow it will wear away too fast. Another objection is that with a narrow seat it will take less scale flaked off to start a leak.

Dividing the seating surface of the valve into a number of concentric rings, a patented process, will lengthen the time between grindings. The lessened area will keep the scale thin, while the grooves will prevent the scale from flaking off across the valve.

As a general rule high sulphur content is associated with low-grade fuel; that is, oils with high asphaltum contents and high mean boiling points. Such a fuel is inherently slow burning, and it may readily be shown that slow burning produces high exhaust temperatures; hence careful regulation of the fuel valves to get the best possible atomization is important.

The Influence of Clearance in compressors is not so detrimental to the efficiency as has often been claimed. It is, however, very detrimental to the capacity of the compressor and the more so the greater the difference between the high and low pressures.

EDITORIALS

F. R. LOW, EDITOR

An Attempt That Failed

THIS year the National Association of Stationary Engineers will hold its convention at Atlantic City. Inasmuch as the meeting is to be devoted entirely to business and social events, supplemented by the usual exhibits of power equipment, the choice of place was most fitting.

Two or three years ago a group of serious minded members undertook to place the annual conventions on a par with those of other national engineering societies by arranging a technical program of papers and addresses by prominent engineers. The idea was most commendable and appeared to offer much for the future of the association. However, in spite of indefatigable efforts on the part of the technical committee, the attempt failed. The papers and addresses were not well received. Two successive conventions gave unmistakable proof that the delegates were not there for technical problems but rather to renew old acquaintances, transact such business as must come before the convention, and have a good time in general.

Recognizing that it is a mistake to try to force anything upon a non-receptive audience, the committee has wisely omitted technical features from this year's program. The educational requirements of the Association now appear to be well taken care of by the "merit system" and the loose-leaf data service, so it is just as well to accept the obvious.

Once more *Power* extends its cordial greetings to the delegates, reiterates its faith in the aims of the Association and wishes it a most enjoyable convention.

Welding in the Universities

ENGINEERING schools are gradually awakening to the fact that the art and science of fusion welding deserve a place in the engineering curriculum. This movement might seem out of step with the present trend toward teaching more fundamentals and fewer specialties, but such is not the case. Welding is now established as one of the fundamentals, with applications as broad as industry itself.

An example will make more clear the distinction between fundamental and specialty. Leaders in engineering education do not consider it wise or necessary to turn out finished boiler designers. Boiler design is a specialty that only a small fraction of the graduates will follow professionally. But all engineering courses devote considerable attention to the design of riveted joints, which are an important factor in boiler design. Every engineering graduate has learned what a riveted joint looks like and knows that it may fail in several ways—by shearing or crushing of the rivets, by failure of the plate ligaments in tension, and so on. If he has attended to his studies, he can figure the efficiency of a lap or butt joint of any of the ordinary types, given a dimensioned drawing and the necessary data as to the strength of the metal in shear, tension and crushing.

Riveted joints and welded joints stand side by side as engineering fundamentals, and they must likewise stand together in the well-balanced curriculum. The graduate must be prepared to consider both possibilities from the first conception of a new project so that he can design *for* riveting or *for* welding. This is a matter of major importance.

Although the broad application of welding in industry is as young as the post-war period, some of the colleges have already made a start in this direction. Others will doubtless follow suit at an early date. The American Welding Society is therefore to be congratulated for its vision in devoting one session at its next fall meeting in Buffalo to a discussion of the place of welding in the engineering curriculum. This session will serve to acquaint the whole fraternity of engineering educators with the good work already being done along this line in a number of colleges. The discussion will be particularly valuable if it crystallizes the best existing thought on effective methods of teaching young engineers rational methods of designing for welding.

Limitations of Boiler Capacity

ENGINEERS are now giving much attention to increased boiler ratings as a means of reducing plant investment, and considerable progress has been made in this direction. It has been found that heat transfer by radiant heat may result in the absorption of 40,000 to 70,000 B.t.u. per square foot of exposed surface which corresponds to about 1,200 to 2,100 per cent of normal rating as generally understood in America. These are probably the upper limits of commercial heat transfer. Boilers are now being installed to operate continuously between five hundred and six hundred per cent rating and some at even higher ratings.

The permissible ratings are largely governed by feedwater characteristics. The foregoing heat transfers by radiant heat can be maintained only with clean tubes and pure feed water. High ratings mean rapid evaporation and concentration in the boilers, so that even with feed water free from scale-forming salts, difficulty would soon be encountered at high ratings, with foaming as a result of the concentration of the soluble salts if these are present in the feed water.

Pure feed water may be obtained in large condensing central stations where condensate and distilled makeup constitute the feed. Care must be taken that condenser leakage is reduced to a minimum; otherwise scale-forming and soluble salts will enter the boiler in excessive quantities.

In many industrial plants much steam may be consumed in the various processes. The resultant large makeup may be too great to be furnished by evap-

orators, hence raw water, after undergoing certain treatment, must be furnished to the boilers. Careful treatment is necessary to remove the scale-forming impurities of natural waters. The soluble impurities cannot be totally removed, and their concentration in boilers entails large blow-down losses to avoid foaming.

Scale formation or high alkalinity in the boiler will reduce the permissible operating rating. Therefore many industrial plants with raw water makeup find it advisable to limit capacities to three hundred per cent rating or less.

These considerations indicate the economic importance of feed-water treatment, which is a serious and individual problem in every industrial plant. Limitations in boiler rating due to impurities in the feed water require a larger and more costly boiler plant than if higher ratings were permissible. Hence industrial plant engineers and chemists must attack the problem of water treatment, where large makeup is required, from the newer viewpoint of limitations of boiler capacity and the consequent expense of additional boiler plant over one with pure feed water.

Testing Regenerative Power Plants

THE high efficiency possible with regenerative power plants, offset by only a relatively small increase in complication, is the reason why practically all large modern power stations are designed for bleeding today. In the last five years this type of plant has grown from the experimental stage to its present prominence.

However, with one or two exceptions, tests of these units have been limited to operation straight condensing; that is, with no bleeding and the heaters bypassed. Why shouldn't the manner of testing keep pace with the improvement in design?

The difficulty in testing on the bleeding basis seems due to lack of foresight in the design of a plant. Now that the operating design features have been quite thoroughly worked out, the next step would logically be to care for some of the lesser details, such as the provision for ample testing facilities. At the present time the large power stations provide sufficient tank capacity to weigh the condensate. So, with a regenerative plant, the mere quantity that must be weighed presents no new difficulty.

There is just one additional factor in determining the throttle flow that requires consideration. The amount of steam bled must be added to the steam exhausted to the main condenser. Generally, surface heaters are used, and where the steam is bled from more than one stage, the complication is reduced by trapping the drip from each heater to the next lower one. As a result, the drip from the lowest pressure heater ordinarily includes all the steam bled. This drip may either be delivered to the condenser or be pumped into the feed line leaving the low-pressure heater. At any point after these drips have been returned to the main feed-water flow, the water can be led to the weighing tanks and the entire throttle flow measured. An additional pump will be necessary to return this water from the weighing tanks at normal pressure to the system. A possible complication in some stations would arise from the fact that the water may be hot enough to cause difficulty due to rapid evaporation in the weighing tanks.

If the temperature of the feed water is too high at a point where all the drips have been added, the water flow as it leaves the high-pressure heater could be cooled to, say, 120 deg. F. and weighed as a whole. This would not affect the performance of the regenerative unit, although it would require an additional heat interchanger and would have some influence on the boiler performance. Whatever method is used, the heaters must operate with their normal water flow at normal temperatures, so that the quantities of steam extracted may represent actual operating conditions.

Two difficulties, not insurmountable, that must be solved are, first, the leakage from the feed-water circuit into the bled steam in the heaters, and, secondly, the complication introduced in the feed-water circuit by oil coolers, generator air coolers, ejector steam condensers, etc., which could be eliminated by bypassing in most plants.

The operator can best protect his own interests by arranging to test his turbines with full bleeding, because of the influence of tests on design. The designer naturally lays out a turbine to show its best performance under the conditions he expects to prevail during the acceptance tests. Hence, the present practice is usually to design the machine to do its best with no bleeding, and in consequence, when steam is bled, the efficiency is somewhat lower than it might be. Since the unit will operate with full bleeding throughout its normal life, the resulting loss to the operator may prove to be considerable. Turbines should be designed for operating conditions and tested under the conditions for which they are designed.

The Good Old Days

THERE are many people who are constantly finding fault with our engineering civilization and are singing the glories of the good old days. These critics of our times find nothing to commend about a civilization which, by releasing an enormous amount of mechanical power to supplement human labor, has produced comfortable modes of living for everybody.

To them the old oaken bucket and the drudgery of those who had to carry water from the stream to their home is preferable to the modern systems by which pure water is pumped by machinery from distant water sources and delivered to our homes. They consider the poet or the artist of old as representing a higher type of manhood than the engineer whose contributions have enabled us to see into the night by electric light, to explore the human body by X-rays, to see the world in moving pictures, to fly through the air like birds, to send the human voice thousands of miles through empty space, and to run faster than the deer by the use of modern power vehicles.

They find little to praise in a civilization in which there is so little difference between the rich and the poor as far as the food they eat, the clothes they wear, the conveniences of their homes, the educational advantages they enjoy, or the general conditions of life which tend to raise all normal minds to a common level of happiness.

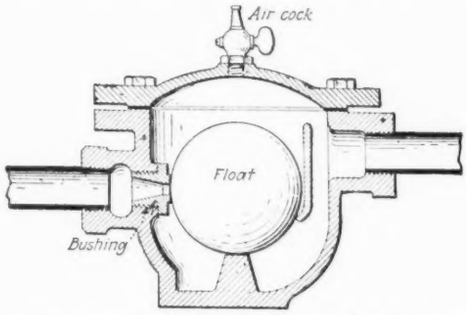
The contrasts of poverty in the good old days may have been more inspiring to the poet, but the lack of inequality among civilized people and their comfortable living are much more conducive to human welfare.

Are we engineers not at fault in keeping silent while the critics of our times who care little about true values are minimizing the true worth of the work of the engineer?

Practical Ideas From Practical Men

Use of Wrong Discharge Bushing Makes Trap Inoperative

The illustration shows the type of trap used on the returns of high-pressure heating coils installed in machines used in process work. The machines have 2,000 sq.ft. of radiation each, using steam at 75 lb. pressure. One of the traps was found to be leaking. This was



Size of discharge bushing is determined by operating pressure on trap

repaired and the bypass valve was renewed, under the supervision of the master mechanic.

The following day it was reported by the foreman that the trap was not working properly. He had to open the bypass in order to keep the coils drained of water. So the engineer was called on the job, to locate the trouble. He found that the outlet bushing, as indicated in the sketch, had been removed and replaced by a new one, but that a low-pressure bushing suitable for 5 to 30 lb. pressure had been installed in place of a high-pressure bushing good for from 75 to 100 lb. When the proper bushing was put in, the trap worked perfectly. As will be apparent from the illustration, the steam pressure at the inlet side of the trap keeps the ball against the bushing until sufficient condensation enters to float the ball away from the bushing, allowing the water to discharge. The area of the low-pressure bushing multiplied by the steam pressure was found to be 46 lb., tending to keep the float against the bushing. With the high-pressure bushing of much smaller area it was found to be only 26 pounds.

New York City. THOMAS SHEEHAN.

Reducing the Generating Capacity Required by Improving the Power Factor

A case was reported to me some time ago of a plant that had to run both its engines when the load could be carried by one if the power factor could be kept above 90 per cent, but as it was frequently as low as 56 per cent, one unit could not hold up the voltage.

This plant had waterwheels located across the river from the steam plant. The steam plant ran all night at light load as the water stopped coming down the river at 5 p.m. due to the stopping of the wheels in a plant about 2 miles upstream. No water was then available till the dam overflowed at the upper plant, and this did not usually occur till about midnight.

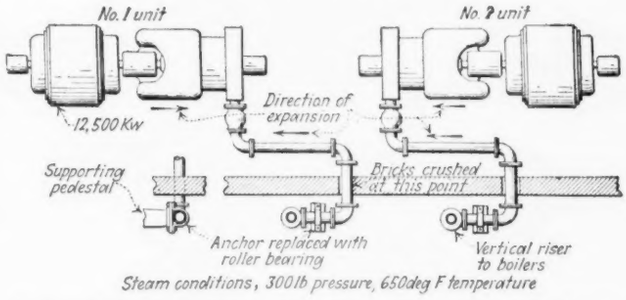
The engineer in the steam plant had to cross the bridge to the hydro plant and shut down the wheels at 5 o'clock or soon after, as they had little storage in their pond. He told me that on stormy nights he always delayed the shutting down till a favorable time, which often emptied the pond and caused the engines to motor the waterwheels. It took about 50 kw. to motor the wheels, but this operation made them function as condensers, for the ammeter on his generator panel on the switchboard at once showed a reduced reading and raised the power factor to a point where one engine could carry the load without any reduction in voltage.

If these wheels were connected to the generator with a clutch instead of being direct-connected, the problem would be simple. All that would be necessary then would be to throw out the clutch and the generator would run as a motor as long as the switches were in, and the wheels could be shut down till water was available. Such arrangements as this are looked upon as too radical in many places, and this plant is still struggling with low power factor. G. H. HART.

Thompsonville, Conn.

Incorrect Steam-Piping Layout Causes Excessive Turbine Vibration

In a certain Mid-Western power plant there were two 12,500-kw. high-pressure turbo alternators of the same type and make, facing each other on the turbine-room floor. For over a year No. 1 unit had given all



Expansion of steam pipe to No. 1 unit caused distortion of turbine body

kinds of trouble, and at times it would vibrate so badly that it had to be taken off the lines, yet No. 2 machine ran perfectly day in and out and in fact it seemed to run itself. No. 1 had to be nursed up to speed, and even then it was hard to keep it in service. The cover had been lifted several times, but there was nothing discovered that would indicate the cause of the trouble. Different engineers had examined the machine to no avail. Finally, one engineer came on the job who attacked the trouble from a different angle. He first

inquired what the steam conditions were and was told that the steam pressure was 300 lb. and the temperature around 650 deg. F. He then took some measurements and found that the overhead valve casing into which the steam entered, twisted on becoming heated by the steam. Upon following the steam line back to where it entered the boiler room, he found the bricks had been jammed and crushed on the side away from the turbine, showing the pipe had been pushing hard that way.

Upon further examination of the piping arrangement he found that the steam pipe from the boiler room to the No. 2 machine entered the turbine room about opposite the middle bearing and ran horizontally from there to the throttle valve and valve casing, and that the pipe to the No. 1 machine entered the turbine room about opposite the valve casing of the No. 2 machine and then ran to the No. 1 throttle and valve casing as shown in the illustration, the difference being that in the No. 2 machine the expansion of the piping was with the expansion of the turbine and on the No. 1 machine the expansion of the piping bucked the expansion of the machine, causing a twisting motion of the valve casing. By releasing an anchor just inside of the boiler room on the line to No. 1 machine and providing a roller bearing for the line at this point, further trouble with balance of No. 1 machine was obviated.

Norfolk, Va. W. A. VANDEGRIFT.

Why Cold Water Pipes Scaled

At the Oscar Mayer Packing Plant considerable trouble has been experienced from the formation of scale in cold-water piping. The conditions no doubt find duplication in other plants and an outline of the solution of the problem may be of benefit to other engineers.

The supply of water is obtained from a well located at the plant, and for the last four years this water has been handled by an air lift. It was found that the interior of the cold-water piping had accumulated a layer of scale at least one inch thick, a four-inch pipe being reduced to an internal diameter of two inches.

The water leaves the well at 52 deg. F. and is run over an ammonia condenser, the temperature rising to 77 deg. F. This water has the same hardness as that of the Madison water works, 293 parts per million.

The study of the cause of this scale brought out the following facts. In water from the limestone region the ions calcium, magnesium and iron and manganese, if present, are present by virtue of the solvent action of carbon dioxide on the carbonates of these ions and, in the case of iron and manganese, of the action of this gas on the oxides, converting them to carbonates and thence into the soluble form through the action of the dissolved CO_2. These ions in the presence of an excess of carbon dioxide are assumed to exist in the form of bicarbonates.

These soluble compounds exist as such only in the absence of oxygen. Thus, if a water containing these salts is pumped out of a well by means of a cylinder pump, the excess carbon dioxide is not released and the compounds remain in solution, owing to their solubility, in an excess of CO_2, which in the presence of water acts like carbonic acid.

Now if the water is pumped to the top of the well and allowed to flow into an open reservoir in the absence of oxygen, the pressure is released and carbon dioxide escapes. If the CO_2 is reduced sufficiently by this lesser pressure, the iron is precipitated and the normal ferrous carbonate is formed as a grayish precipitate. This precipitate, however, is formed only in the *absence* of oxygen.

If the water is pumped into an open reservoir in the presence of oxygen, the ferrous carbonate is oxidized and precipitated as ferric hydroxide, which is one of the chief constituents of the reddish brown deposits, known as rust. If this hydroxide is dried, the compound loses water and the ferric oxide is formed. If the oxidation is not complete and the drying is accomplished, then there will be some ferrous oxide mixed with the ferric hydroxide, and in this case the compound is designated by the formula Fe_3O_4.

Thus it is shown that the presence of carbon dioxide in excess holds iron in solution and the liberation of this excess in the presence of air causes the formation of the iron in the insoluble form, as a reddish brown deposit.

Calcium and magnesium are also in a like manner precipitated from the water solution by the liberation and expulsion of carbon dioxide. In the case of these ions, however, oxygen plays no part as an oxidant. The deposition of calcium and magnesium from their solution as bicarbonates, depends upon any process that will remove carbon dioxide. They are not removed so readily by simply allowing the excess carbon dioxide to escape from the water in contact with air.

In the cylinder pump system the carbon dioxide is retained in the water by virtue of the pressure, a gas being soluble in water in direct proportion to the pressure. If the water is maintained under pressure without admission of air, no appreciable change occurs in the water.

In the air-lift system, the air serves to disturb the equilibrium and carbon dioxide is released and driven off with the air, which is low in carbon dioxide. The tendency of this system is to alter the equilibrium so that aëration actually removes iron from the soluble condition to the insoluble ferric oxide. The aëration also alters the stability of the calcium compound and reduces it to the minimal condition of stability, which is easily changed to the unstable condition by slight rises of temperature.

The condition obtaining before the installation of the air-lift pump prevented the existence of minimum stability, and subsequent heating by the coils only tended to produce minimal stability and very little carbonate of lime was deposited through heating of the water from 52 to 77 deg. F. The air-lift pump provided the condition of minimum stability to the water and then the slight rise in temperature causes the rapid deposition of the lime in the pipes. There are two remedies for this calcium carbonate and oxide of iron deposit in the pipes: (a) Replace the present air-lift with a centrifugal pump; (b) take the water for the plant supply directly from the well before it has been raised in temperature by being allowed to run over the ammonia condenser and use the present air lift.

To determine which solution or remedy is the more economical, it is necessary to make an analysis of each individual remedy. The writer wishes to mention with special gratitude the assistance of M. Starr Nichols, chemist and assistant bacteriologist of the State Laboratory of Hygiene of Wisconsin.

RONALD G. REYNOLDSON, Mechanical Engineer,
Madison, Wis. OSCAR MAYER & Co.

Comments from Readers

How the Discharge Pressure of a Centrifugal Pump Was Increased

On page 1017 of the June 29 issue R. C. Whitehead shows how he changed the suction line of a centrifugal pump to increase the discharge pressure. It seems to me that there was a more practical way of overcoming the trouble than the one used.

Undoubtedly, a great deal of friction was eliminated by changing the suction piping as described, and shown by the illustrations reproduced herewith, but it is not reasonable to expect that the entire gain in pressure can be attributed to the short suction pipe and elbow alone. No doubt the original suction pipe was too close to the bottom of the sump, and there was considerable throttling action at this point, which was eliminated when the piping was changed. The velocity of the water in the approach channel was also high, indicating an avoidable loss of head of about one-third of a foot.

When the piping was changed to take advantage of the velocity of the water, the suction lift was further decreased, and it is doubtful if more than a fraction of a foot can be charged to friction in the original piping proper.

A better method in my opinion, would have been to shorten the original piping and take the check out of

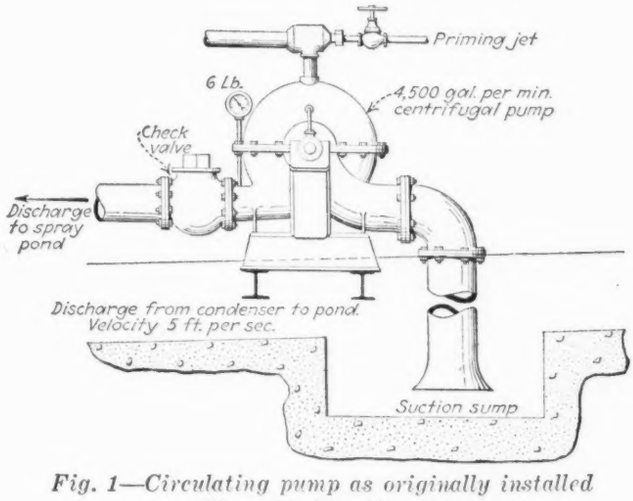

Priming jet
6 Lb.
4,500 gal per min. centrifugal pump
Check valve
Discharge to spray pond
Discharge from condenser to pond. Velocity 5 ft. per sec.
Suction sump

Fig. 1—Circulating pump as originally installed with vertical suction pipe

the discharge and substitute a gate valve. A check is a considerable source of loss of head, and there is no good reason for installing it on a low-lift pump under such conditions. In fact, there is every reason why it should not be installed in preference to a gate valve. The friction head required to hold the check open is a large percentage of the total head on a low-lift pump and is avoidable. Usually, standard checks are used and the engineer does not think to lighten the check as he should do. Furthermore, the pump must be started against little or no head and requires an excessive starting current. A centrifugal pump should be started up with a closed discharge to keep the power requirements at a minimum.

If the pump should stop in an installation such as described, no damage would result, as would be the case in a high-head installation with a large volume of water in the discharge piping.

The head on the spray nozzles could have been boosted by cutting out some of the sprays. In any event the

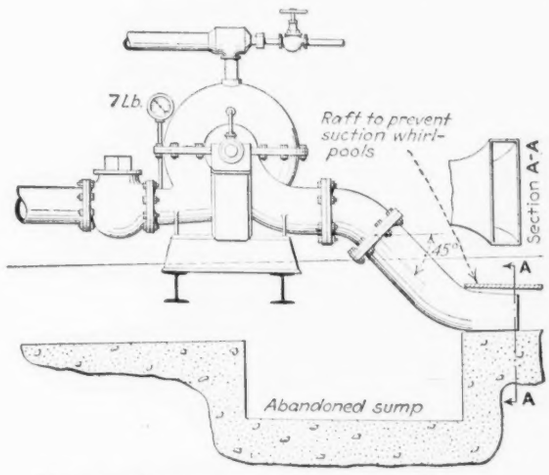


Fig. 2—Change made in suction pipe

difference in the cooling effect between a 6- and a 7-lb. pressure would not have been great enough to cause any great concern unless the plant was operating on the ragged edge. In my opinion the installation shows rather poor design throughout. B. MARKER.

Los Angeles, Calif.

"Breaking Into the Game"

The experiences with a new engineer described by O. S. Marugg in the June 8 issue will be doubted by some who have not come across such cases. It is for the purpose of assuring them that Mr. Marugg's is not an isolated case that I relate my experience. Incidentally, it may answer Mr. Marugg's question, "How do they do it?"

Early in 1920 I was called into my superior's office and introduced to a middle-aged man, Mr. X. I was told that Mr. X had joined the engineering staff of the company and that, for the time being, he was to act as my assistant.

Later in the day my superior again called me to his office and informed me that Mr. X was a "gold seal" licensed man, that he had served as an officer of high rank in the army and had had exceptional experience. "He is just the man we want for our new 10,000-kw. turbine, and we cannot afford to pass him up. He is willing to work for a low salary until we have an opening for him." Mr. X proved to be a man devoid of any engineering training or experience.

After roaming around the plant a few days, he began

to ask questions of unskilled help. He asked a Portuguese laborer who was knocking scale out of the header caps of a water-tube boiler, whether the caps were cast in the company's own foundry and what part of the boiler they fitted on. He asked the water tender what the "sticks at the water gage" were for. When the water tender asked him if he was referring to the try-cock handles, he answered, "Yes, the try-cock handles; what are try-cocks for? It is a long time since I operated a boiler; I have forgotten."

Because of his prolific talking he became an object of ridicule of the entire staff, notwithstanding his age, his "gold seal" license, and the high officer's title prefixed to his name.

When the true caliber of the man became known to my superior, he asked me how Mr. X was measuring up to the duties assigned to him, and when I hesitated to answer, he said: "I am sorry. I made a mistake in taking him on. But there must be something he is good for, something that he can do. I leave it to you to make some use of him."

I must admit that I failed, notwithstanding my experience in training apprentices in vocational schools and in the field. I did my best to get him to do the simplest task, such as weighing coal on a boiler test, but even that wasn't done right, according to the coal passer. The results of the test proved that.

Asked how he happened to get the highest-grade license, he said that through friendship and because of his age and an enlistment in his younger days, he received a high-rank commission in the engineering corps. He was assigned to take charge of a power plant in an ammunition factory. That post required the highest-grade license, for which he applied. He appeared before the examiners in full uniform, shook hands and walked out with a "gold seal" license!

"Veni, vidi, vici." Julius Cæsar had nothing on him!

New York City. E. H. PHILLIPS.

Economy of High-Pressure Steam in Industrial Plants

The opinion prevails that the cost of high-pressure equipment prohibits its use in industrial plants. This is not the real reason, because if the owners of such plants realized the saving they could effect, plants from 2,000 to 3,000 kw. capacity would be built with higher pressures than those used now.

In a high-pressure installation it is well to consider how the steam can be used, because to use 600 lb. pressure in a compound- or multiple-expansion reciprocating engine, the cylinder condensation would offset any saving. If the steam is superheated, the cylinder lubrication becomes a problem.

The turbine is more suitable, but the same difficulty with condensation in the lower stages will be encountered even if the steam is superheated, therefore, a high-pressure unit, independent of the lower-pressure unit, should be used. Assume a plant of 2,000 kw. wished to increase its capacity to 3,000 kw. There are two ways to do this—to rebuild the whole or to enlarge it. The latter is usually done because it is cheaper.

The addition should consist of high-pressure steam-generating equipment having a capacity of 30,000 lb. per hour, around 600 lb. gage; a 500-kw. turbine taking steam at 600 lb. and exhausting at the pressure of the old installation; precipitators might also be installed on the top of the old boilers. Although the use of precipitators is not new, the way they are used now is new; that is, live steam from the boiler goes direct to a drum with baffle plates arranged so that the steam mixes with the feed water. The purpose of this arrangement is to precipitate the solids in the feed water and to facilitate a better circulation in the boiler. To use this installation, non-return valves are placed in the proper location. Either the exhaust steam of the high-pressure turbine or live steam from the old boilers is mixing with the feed water.

Assume the steam rate of the high-pressure turbine to be 60 lb. per kw.-hr., this would make 30,000 lb. of steam available to the old installation which has a steam rate of 20 lb. per kw.-hr. or a total hourly steam rate of 40,000 lb. By installing a new prime mover of 500 kw. capacity working under the same condition as the old ones, the total steam rate would be increased to 50,000 lb. of steam. Then we would have 50,000 lb. of steam generating 3,000 kw.-hr., where 40,000 lb. generated 2,000 kw.-hr. before, or an equivalent saving of 10,000 lb. of steam an hour.

Reliability is assured because the original plant is the same as it was before, and in addition, 20,000 lb. of steam-generating capacity is available as spare or for the future power demand. OTTO QUENZEL.

Chicago, Ill.

Are Welded Pressure Vessels Safe?

I am interested in the subject, "Are Welded Pressure Vessels Safe?" which has been under discussion in your columns as well as in numerous other engineering publications. Perhaps my interest is made somewhat keener because for several years I have been in charge of inspections on a large number of such objects, principally air tanks. I have been impressed with the fact that air tanks are elements of danger to a far greater degree than is realized by most users and many engineers. I have expressed this opinion before in your columns for the purpose of arousing interest in this much neglected subject, and it is pleasing to note that unfired pressure vessels are now coming in for their share of attention, even if brought about largely by the question of whether or not they should be constructed by the welding process.

S. W. Miller, of the Union Carbide and Carbon Research Laboratories, Inc., seems to hold the floor in the interest of autogenous welding of pressure vessels. I have read carefully his letter in *Power* of June 29 and his article in *Mechanical Engineering* of several months ago, as well as many more of his writings and sayings on the subject. One cannot but be impressed by Mr. Miller's reputation as an engineer and his knowledge of the art of welding. But one is also impressed with the fact that he is talking for his product.

In his article in *Mechanical Engineering*, Mr. Miller criticizes the Boiler Code Committee of the American Society of Mechanical Engineers for its formulation of that section of the code relating to construction of unfired pressure vessels. He took the position that the society should not produce codes that deal with engineering matters from a public safety point of view, but that its codes should consist of specifications based on the very latest engineering methods, without attempting to restrict the general use of such methods by everyone, by injecting a safety factor sufficient to compensate for

poor application of method. Accordingly, he proposes a revised code that carries his ideas and that, if adopted, would give welded tanks an equal standing with those of riveted construction.

I disagree with Mr. Miller as to the proper character of a boiler construction code. One motive behind the Boiler Code was to produce a standard of boiler construction and equipment that could be adopted by all local authorities throughout the United States and Canada as a boiler law, thus eliminating the confusion and economic waste hitherto experienced in having to design boilers in any one shop to meet varying requirements of various political subdivisions. Hence such a code must take into consideration not only possible laboratory methods, but the probable methods of a large variety of prospective manufacturers and the general run of so-called skilled labor. It cannot properly set up as a standard the ultimate in scientific achievement, leaving for the various law-enforcing bodies the very difficult problem of compelling the average skilled workman to turn out every job with the same scientific exactness as the proponents of welded tanks have been able to do with their test specimens.

The art of autogenous welding is still far from perfect and will not become an exact science until it can be accomplished without materially altering the chemical and physical characteristics of the metal fused as well as changing the stresses adjacent thereto. I have seen dozens of welded air tanks that exploded under normal working pressures from 200 down to 40 lb. per square inch. In every case the initial rupture was traced to a weakness produced by the act of welding, usually embrittlement, sometimes insufficient fusing. Of the several riveted air tanks I have seen after explosion at normal working pressure, none failed in the seam. The usual cause has been internal corrosion. The only riveted-tank explosion I recall wherein failure was in the seam was on a tank that for a long period had been operated at a pressure giving a factor of safety of 2½. And it should be kept in mind that there has not been the same care used in air-tank riveting generally as is the case in boiler construction, owing principally to the fact that air-tank construction is not usually subject to the same rigid shop inspection requirements.

In answer to Mr. Miller's attempted proof that Mr. Gillmor's argument in the Feb. 9 issue is fallacious, I would point out:

First: There is not "plenty of good welding" being done as long as there is so much poor welding. And there are plenty of so-called responsible manufacturers who have turned out and are turning out pressure vessels that are a menace to human life.

Second: It is a simple matter to detect poor workmanship in riveted construction during the progress of the work. One need not be a metallurgist to drill holes and drive rivets or to discern when such operations are being improperly done. And therein lies the great advantage of riveted construction of pressure vessels.

Many manufacturers have stated that in a shop fully equipped for both there is practically no difference in cost of construction as between welding and riveting except in the smaller sizes. Even if riveting is costlier, the difference is a worth-while investment in safety.

It has taken us many years to get away from old hand-made boiler-shop practices. The boiler situation, as far as new construction is concerned, is excellent. The explosion of a Code boiler due to an inherent defect either of material or workmanship is practically unknown. I wonder if that could be said if for the past ten years we had been building them with welded longitudinal seams and the best present-day autogenous welding methods.

A. J. LAMIE.

San Francisco, Calif.

Engineering and the Man

I have been following with great interest the discussion in *Power* concerning the number of students dropping out of engineering courses before completion.

I quit engineering after eleven years' practice to take charge of a business where engineering experience is valuable but not indispensable.

After being out of college ten years, the secretary of our class—a large class, too—compiled statistics on earned income in a confidential manner, insuring privacy for the individual return. These statistics largely influenced me in changing from engineering to business. The list follows:

		Average Income
10	bankers and brokers	$11,040
36	manufacturers	8,524
18	physicians	7,944
7	architects	7,230
31	merchants	7,136
29	lawyers	5,905
41	salesmen	5,690
3	purchasing agents	5,600
4	advertising	5,476
5	newspaper	5,180
5	accounts and statistics	5,004
6	insurance	4,533
14	farmers	4,461
15	builders and contractors	4,287
7	veterinarians	3,921
83	engineers	3,724
33	teachers	3,137

The actual incomes are not applicable today, as this table is several years old, but the relative incomes of the different vocations are illuminating.

On this basis, would one be wise in having a son study engineering, if that son had to earn his living in the profession in which he was trained?

New York City. C. G. HOWARD.

Nothing New Under the Sun

I do not wish to rob Machinist Mate Waldron of any credit in the work of raising the "S-51," but the nozzle that he designed is not exactly a new idea, that is, as far as this section of the country is concerned.

Several years ago we used a nozzle of this kind made out of pipe fittings screwed onto a 1½-in. pipe to clear a 16-in. pipe that was plugged up with ashes for about 100 ft. The jet from the front orifice loosened the ashes, and streams issuing from the backholes washed them down, as well as helped to propel the nozzle forward.

We have used a smaller nozzle of this type for several years to blow out the fine coal and ashes from the air boxes of our stokers; in this case we use air.

Wilkesbarre, Pa. THOMAS M. STREET.

THE EFFECTIVE USE of the flue-gas air heater is favored by the general adoption of stage feed-water heating by bled steam. The resulting increased feed temperature reduces the opportunity for heat recovery in an economizer, but in no way limits the air heater. It should not, however, be considered that economizers are a dead issue. They are being installed in some of the largest and newest plants. A good example is the recently opened Avon plant of the Cleveland Electric Illuminating Company.

Questions and Answers

Lubricating Oil Consumption of Diesel Engines

What is the lubricating oil consumption of a Mietz & Weiss semi-Diesel engine of 50 hp. rating, operating at 340 r.p.m.? What is the fuel consumption that one might expect from this engine at full load and ¾ load?

With this type of engine, which has a very low compression, the lubricating oil used should be not more than one gallon in 10 hours. In other words, the engine should deliver 500 hp.-hr. at full load with a consumption of one gallon of lubricating oil. The fuel-oil consumption of this engine is somewhat higher than that of engines of greater compression pressures, and one should be satisfied to obtain 8 hp.-hr. per gal. of fuel oil at full load and about 7½ hp.-hr. at ¾ load. In other words, when carrying 50 hp., this engine will use about 6¼ gal. of fuel oil per hour, and when carrying 37½ hp. the engine will use about 5 gal. per hour.

Size of Belt to Drive Press

We have a 15-hp., 1,750-r.p.m. electric motor which drives a large press. The motor is belted to a countershaft, the center distance between the motor shaft and the countershaft being 15 ft. The pulley on the motor is 6 in. in diameter and the countershaft pulley, 40 in. in diameter. What width of leather belt would you advise and should it be a single or a double belt?

A belt drive, under good conditions of operation, can be designed for an effective tension of 55 lb. per in. of width in case a good grade of single leather belt is used. Your conditions of service could not be characterized as "good" because the diameter of the small pulley is quite small and the ratio of pulley diameters is large. Choosing an effective tension of 45 lb. per in. of width should compensate for these unfavorable conditions.

Using this effective tension, the belt size required for your service is 4 inches, found as follows: The surface speed of the small pulley which may be taken as the belt speed is

$$\frac{\pi \times 6}{12} \times 1{,}750 = 2{,}750 \text{ ft. per min.}$$

The width of belt required to transmit 15 hp. is

$$w = \frac{33{,}000 \times hp.}{S \times E} = \frac{33{,}000 \times 15}{2{,}750 \times 45} = 4 \text{ inches.}$$

Where $hp.$ = horsepower transmitted,
S = belt speed in feet per minute.
E = effective tension in pounds per inch of width.

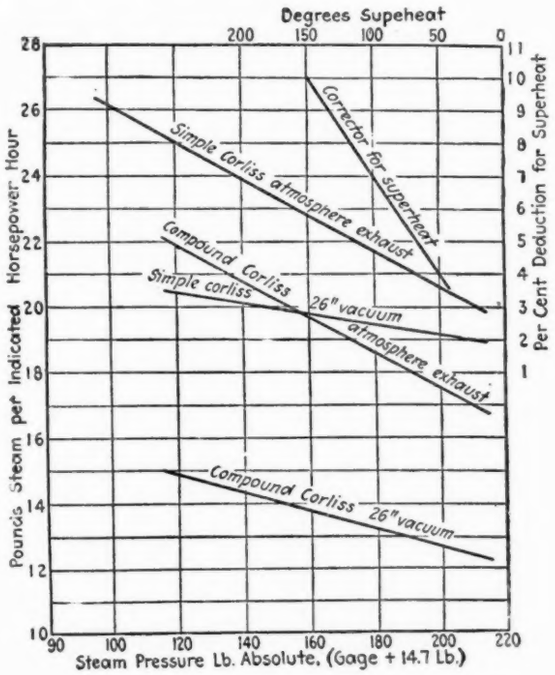
A single belt is better suited than a double one because of the small diameter of the motor pulley and the small power transmitted.

There is one other point that may or may not merit consideration. You did not specify the type of press. If this is a punch press, it may be that the motor is overloaded during the short period of operation. The motor would stand such service but, if you suspected such a condition to exist, it might be necessary to specify a larger belt. The characteristics of a leather belt are such, however, as to permit considerable overloading if properly installed.

Effect of Higher Pressures on Steam Consumption

We have a 20x36-in. Corliss engine direct-connected to a direct-current 250-volt 1,000-amp. 100-r.p.m. generator. With a load of 600 amperes and with 80 to 90 lb. steam pressure and 2 lb. back pressure, the engine runs smoothly. If the steam pressure is raised to 120 lb., with the same load and back pressure, the engine does not run so smoothly. How much will the steam consumption be reduced per kilowatt-hour by operating at 120 lb. pressure, and show by example how you calculate this? Also, please calculate the steam consumption per kilowatt-hour from an indicator diagram.

It is difficult to calculate the steam consumption from an indicator diagram within 25 per cent of accuracy. The only way this can be done is by a test,



Steam consumption of various engines

measuring the steam consumed and the power generated. In the figure are shown steam rates of typical engines. With steam at 105 lb. absolute (90 lb. gage) the steam rate per indicated horsepower-hour for a Corliss engine with atmospheric exhaust is 26 lb., which is 41.9 lb. per kw.-hr., if we assume the engine's mechanical efficiency to be 90 per cent and the generator efficiency 92 per cent. Increasing the pressure to 135 lb. absolute (120 lb. gage) decreases the steam consumption to approximately 24.25 lb. per i.hp.-hr., or 39.1 lb. per kw.-hr. The decrease is then 2.8 lb. per kw.-hr. or $2.8 \times 132 = 369.6$ lb. per hour with load of 132 kw. Inasmuch as the engine is now cutting off at about ¼ stroke, raising the steam pressure will shorten the cutoff, which should further decrease the steam rate below that calculated, by at least 5 per cent.

News in the Field of Power

Shoals Nitrate Impossible—Curtis, Yale Chemist

There is no known method by which the cheap power generated at Muscle Shoals by public funds can be used to manufacture cheap fertilizers, according to Dr. Harry A. Curtis, professor of chemical engineering, Yale University, and one of the five experts in President Coolidge's "Muscle Shoals inquiry" last year.

Asserting flatly that "the cheap power of Muscle Shoals cannot be used to manufacture cheap ammonia-nitrate fertilizer," Dr. Curtis urged instead that the gigantic power supply there be used to manufacture other chemicals than nitrates, which would be of equal or greater value to the American nation.

Dr. Curtis is consultant for the United States fixed nitrogen laboratories, Washington, D. C. During the war he was active as a member of the nitrate division, the United States War Department.

Speaking to a representative of *The Christian Science Monitor*, he said that no man versed in the natural sciences would venture to say that some method of manufacturing cheap fertilizers at Muscle Shoals will not be discovered, but at present, despite popular belief to the contrary, no such method exists.

Other Chemicals Possible

Interest attaches to Dr. Curtis' statement, in light of the certainty that the whole problem of Muscle Shoals will be threshed out again when Congress reassembles in Washington in December.

Summarizing his points, Dr. Curtis said:

"1. Cheap power of Muscle Shoals cannot be used to manufacture cheap fertilizer.

"2. If the power from Muscle Shoals is distributed for general public utility service, the great advantage of a low initial cost will be dissipated, and will not benefit the consumer.

"3. Muscle Shoals can still be dedicated to national defense by manufacturing there those chemical products which are as fully as indispensable as nitrates, but which, unlike nitrates, can be manufactured only by means of cheaper power."

"There are other chemical substances just as vital as nitrates which can be produced economically only with the means of cheap electrical power. Some of these are caustic chloride, ferro-alloys, and abrasives.

"Therefore, it can still be dedicated to national defense, using it for the manufacture of these other chemical products. This would result in building up at Muscle Shoals a group of industries similar to those at Niagara Falls, and would add enormously to the preparedness of the United States.

"The American farmer would not be injured by this program, since a cheap fertilizer product is impossible."

Plan to Save Niagara Falls Scenic Value

Acting as the technical representative of the United States, Major General Edgar Jadwin, recently conferred with Colonel O. M. Biggar, the Canadian representative, in regard to the scope of the work proposed to prevent further recession of the crest at Niagara Falls. The conference was tentative in character.

The United States proposed that this work be undertaken solely with the view of preserving the scenic beauty of the falls. Canada, in reply, suggested that the conference consider also an increase in the diversion of water for power purposes. In the meantime, however, the urgent need which did exist on the Canadian side for additional water power has been met otherwise. This leads to the hope that early steps may be taken to check recession without opening up the highly controverted question of the division between the two countries of the water diverted for power purposes.

While in Canada, General Jadwin also attended a meeting of the Joint Engineering Board which is writing a report on the cost of some of the principal structures which would have to be built in the development of the power resources of the St. Lawrence. Rapid progress has been made on the report, General Jadwin reports.

Exhibits and Entertainment at N.A.S.E. Convention

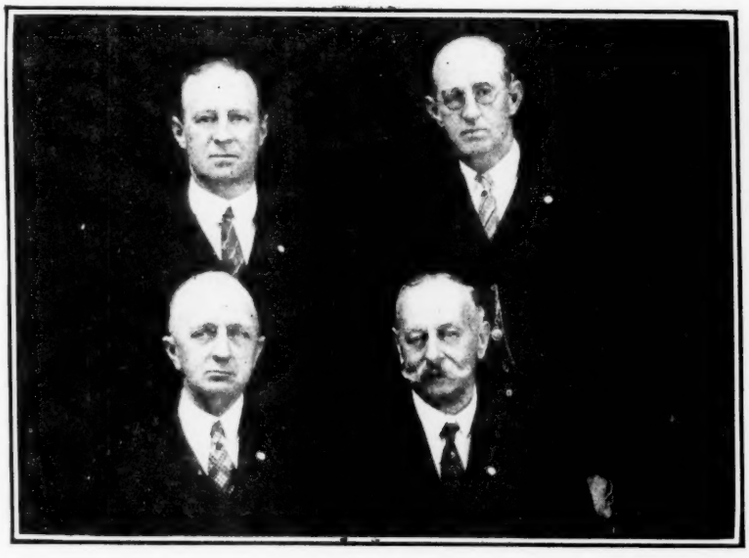
Many booths exhibiting manufacturers' products in the line of machinery and equipment in the power-plant field, together with a number of business meetings, will constitute the leading features of the annual convention of the National Association of Stationary Engineers, to be held this year Sept. 14-17 at Atlantic City, N. J.

The Steel Pier, noted as one of the finest amusement centers in the world, has been secured for the meeting and the housing of the display booths. The exhibition hall adjoins the committee and meeting rooms, on the entrance side of the pier, so facilitating an examination before the assembly rooms are reached. As this issue goes to press, it is reported that most of the booths have already been engaged by exhibitors.

Arrangements by the convention committee make provision for a rolling chair parade for ladies attending Wednesday afternoon.

Entertainment rather than professional considerations will be the order of the convention, as no technical papers will be presented. Music will be rendered each night in the exhibit hall, and the program calls for varied amusement for all.

Those constituting the committee of arrangements are: George Hassan, chairman; John Rutherford, assistant chairman; Thomas Small, secretary; George Ehrmann, assistant secretary; George Armitage, treasurer, and George W. Armitage, assistant treasurer.



N.A.S.E. Convention Committee: seated left to right, Thomas Small and George Armitage—standing, John Rutherford and George Hassan.

Washington Company Plans Large Development

Preliminary steps have been taken for the development of the $20,000,000 hydro-electric project of the Grays Harbor Railway & Light Co. on the Cowlitz River near Mayfield, Washington, according to W. W. Briggs, manager of the company. Five crews are engaged in making test drills on several prospective sites, one hole now being 250 feet deep.

Probably two dams will be constructed, the main dam to be in the vicinity of Mayfield and a second dam about nine miles above that point. This will create an impounding basin to insure an adequate supply at all times for the main basin at Mayfield.

The height of the dams will be determined largely upon the power contracts that can be obtained before actual construction starts. Several corporations are now in negotiations with the company, but the size of the plants they will build is not settled. The first development plans call for the construction of an $8,000,000 plant.

Power Use of Coal and Gas Rises—Oil Use Declines

Average daily consumption of coal and natural gas in the production of electric power by public utility power plants in the United States increased during June as compared with the corresponding period during the last seven years, while consumption of oil declined, according to figures compiled by the Division of Power Resources of the Geological Survey, Department of the Interior.

The average daily consumption of coal in the power plants in June was 105,700 net tons. Corresponding figures for the last seven years were: 1919, 88,300 net tons; 1920, 95,000; 1921, 81,400; 1922, 82,900; 1923, 98,400; 1924, 89,300, and 1925, 103,000.

The plants' average daily consumption of natural gas in June was 142,000,000 cu.ft. Corresponding figures for the last seven years were: 1919, 66,200,000; 1920, 75,200,000; 1921, 69,000,000; 1922, 78,200,000; 1923, 94,400,000; 1924, 142,800,000, and 1925, 125,400,000.

Average daily consumption of oil was 20,700 bbl. Corresponding figures for the last seven years were: 1919, 29,000 bbl.; 1920, 35,900; 1921, 30,500; 1922, 30,700; 1923, 34,500; 1924, 44,800, and in 1925, 26,400.

Cheap Shoals Power Opposed by Northern Mills

In an effort to block the distribution of Muscle Shoals power throughout the South the opponents of the offer of the thirteen southern power companies are reported as circularizing New England, urging that the bill recommended by the Joint Congressional Committee be defeated if that section wishes to stop the migration of the textile industry to the South.

If the general distribution of the power can be prevented, it is pointed out, it then will be devoted to the manufacture of fertilizers and chemicals and will not make cheap power available to cotton mills and other forms of manufacture which compete with New England. That section will profit if the maximum amount of fertilizer is made, since that would make possible the growing of more cotton, which means cheaper cotton for the textile mills.

The inspiration for this propaganda is said to come from those most interested in developing the immediate locality of Muscle Shoals. If all of the power can be consumed near the Wilson Dam it will assure the creation of a great industrial community there.

Reports Vast Water Power in British Columbia

There are vast water-power possibilities in the Kootenay region, which includes the Canadian territory lying west of the main range of the Rocky Mountains, from the headwaters of the Canoe River on the north, southeasterly to the middle latitudes of Montana and Idaho, according to a report from Consul Norton F. Brand, Fernie, British Columbia, made public by the Department of Commerce.

On the Pend d'Oreille River it is believed that 200,000 hp. can be developed and on the lower Kootenay, between Nelson and the confluence with the Columbia, there may be 180,000 more, the report states.

Mining Company Installs Large Diesel Engines

Those who have followed the growth of the Diesel engine in the last few years are fully aware of the tendency toward the use of larger units. This is further evidenced by an order just received by the Nordberg Manufacturing Co., Milwaukee, Wis., from the Commerce Mining & Royalty Company of Miami, Oklahoma, for three 2,250 hp. units, or a total installation of 6,750 hp.

This mining company which is one of the world's largest producers of lead and zinc, has its operations principally in Ottawa County in the northeast corner of Oklahoma and in Cherokee County in the southeast corner of Kansas.

At present a portion of the power is generated by steam and gas engine units located at widely separated points. The major portion of the power, however, is purchased, but when installed, the three Diesel units will comprise the only source of power.

It is planned to locate the Diesel plant at Cardin, Oklahoma. This new plant will be laid out with the thought of future expansion to permit of the installation of additional units should power demands later require it. Each engine, which is of the 4-cylinder two-stroke-cycle type, will be connected to a 2,000-kva., 2300-volt, 3-phase, 25-cycle Westinghouse generator. A large oil storage capacity will be provided to permit the purchase of oil under most favorable conditions.

American Industry Congress To Review Progress

Papers, discussions and exhibits illustrative of the industrial progress of the United States will form the business of the Congress of American Industry to be held in the Commercial Museum, 34th and Spruce Sts., Philadelphia, Pa., from Sept. 7 to 27.

Each day two sessions will be held at which addresses on the various stages of technical, industrial and commercial development will be delivered by a leading authority in each field. Topical papers will be presented but not read at the sessions and brief discussions of matters germane to the subjects will follow the addresses.

Of special interest to engineers are the addresses on "The Engineer," by Dr. Dexter S. Kimball, Cornell University, Sept. 10; "Power and Its Distribution" by Guy E. Tripp, Westinghouse Electric & Manufacturing Co., and "Illumination" by Dr. William C. L. Eglin, of the Franklin Institute.

The exhibit will be in ten distinct episodes, each illustrating a period of fifteen years. The various phases of industry will be introduced as they became part of the life of the country.

Obituary

Andrew Kidd, who for 44 years served as chief engineer for the Burden Iron Co., Troy, N. Y., and was retired April 15, 1926, after 59 years of continuous service in the same plant, died Aug. 22 following an intestinal operation at a Milwaukee hospital.

Until a month ago Mr. Kidd was vigorous alike in mind and body, although at that time he had passed 83 years of age. Born at Lochwinnoch, Scotland, this veteran engineer sailed for America in 1867. The same year he began work for the Burden Iron Co., with his first job that of running a hoisting engine. He was made second engineer in 1873. From 1882 until the spring of this year he continued at the helm as chief engineer, and, probably, with the unique distinction of having operated throughout the whole period of his engineering career the same four vertical Corliss engines installed in 1862 and still in daily operation. *Power*, in the Aug. 1 issue, tells of this in "Operated Same Engines for 59 Years."

Funeral services were held from the residence in Troy, N. Y., on Aug. 25.

Personal Mention

Samuel Insull, Jr., on Aug. 23, was elected to the presidency of the Midland Utilities Co. He is succeeding his father, who, on the same day, became chairman of the board of directors. It will be recalled that the principal subsidiaries of the company are the Northern Indiana Public Service Co., Chicago, South Shore & South Bend Railroad, Gary Railways Co., Indiana Service Corp. and the West Ohio Gas Co.

C. L. Hudson, formerly associated with the Merco Nordstrom Valve Co. at the Houston office, is no longer with the company.

E. C. Williams, formerly assistant electrical engineer for the Public Service Commission of Northern Illinois, has been appointed electrical engineer to succeed Frank E. Goodnow, as announced by J. L. Hecht, vice-president.

Edwin A. Knowlton has established quarters at 516 Rice Bldg., 10 High St., Boston, Mass., as manufacturers' agent for the Klipfel Manufacturing Co., Kelly Valve Co., Michigan Valve & Foundry Co., the L. B. Safety Stop and the Certrifix Corporation.

L. E. Fischer, of St. Louis, Mo., transportation and public utility executive and engineer, has been appointed vice-president in charge of operation of the North American Light & Power Co., according to an announcement by Clement Studebaker, Jr., president of the company.

Harold M. Graham has joined the Ross Heater & Manufacturing Co., of Buffalo, N. Y., as chief engineer. Mr. Graham, active for many years in the condenser departments of the Westinghouse Electric & Manufacturing Co. and the Elliott Co., has done much original work and investigation on the problems of heat transmission and vacuum.

Business Notes

The Westinghouse Electric International Co. has recently made known the appointment of H. H. Rogge, electrical engineer, as special representative to the Philippine Islands. Mr. Rogge's territory will also include the Dutch East Indies, the Malay Peninsula and Siam.

The Harnischfeger Corp., Milwaukee, Wis., builder of overhead electric cranes and gasoline driven excavators, announces that L. N. Ridenour, formerly with Wellman, Seaver, Morgan Co. and later agent for Contractor's Equipment in Cleveland, has been appointed special factory representative.

The Reading Iron Co., Reading, Pa., manufacturer of wrought-iron pipe, charcoal-iron boiler tubes, engine-bolt iron, etc., is represented in Kansas City, Mo., by O. R. Lane, who lately covered the St. Louis territory for the company. Mr. Lane opened an office for the company at 721 Pioneer Trust Bldg., Kansas City.

The Ross Heater & Manufacturing Co., Inc., of Buffalo, announces that Ross H. Rathbun has joined its organization as sales manager. Mr. Rathbun leaves the Westinghouse Electric & Manufacturing Co. with which he has been associated since 1916, when he returned to the United States from Mexico, where a number of years had been spent as chief engineer for the Madera Co., Ltd.

The Beaumont Manufacturing Co. manufacturer of ash hoppers and gates, has appointed L. E. Striegel as district sales engineer of the New York territory. Mr. Striegel, who was formerly chief engineer of the Baker-Dunbar Co., will be located at 50 Church St., New York.

The Chain Belt Co., manufacturer of Rex chains and conveyors, announces the appointment of George E. Coursey and E. E. Elsey as its special representatives in southern Ohio, southeastern Indiana and eastern Kentucky. The firm of George E. Coursey is located at 701 Union Central Bldg., Cincinnati, Ohio.

Coming Conventions

American Society of Mechanical Engineers. Calvin W. Rice, 29 West 39th St., New York City. Regional meeting at Richmond, Va., Sept. 27-30.

Association of Edison Illuminating Companies. Chateau Frontenac, Quebec, Sept. 27-Oct. 1. P. S. Millar, 80th St. and East End Ave., New York.

Empire State Gas and Electric Association. Lake Placid Club, N. Y., Sept. 30-Oct. 1. C. H. B. Chapin, Grand Central Terminal, New York.

National Association of Stationary Engineers. F. W. Raven, 417 South Dearborn St., Chicago, Ill. Convention at Atlantic City, begins Sept. 14-17. Erroneously announced as Sept. 6-11. Annual conventions and exhibitions of state associations are scheduled for the following dates: **New York State Association.** Wilmer T. Meinzer, 3626-215 Place, Bayside, Long Island, N. Y. Postponed from June 11 to 12, at Troy, N. Y., to Sept. 12-13 at Atlantic City, N. J. **New Jersey State Association,** S. G. Dalrymple, 111 Hutton St., Jersey City, Convention at Atlantic City, Sept. 12; **Pennsylvania State Association,** Frank J. McCarron, 3647 North 11th St., Philadelphia, Convention at Philadelphia, Sept. 12-13.

New England Water Works Association, Boston, Mass.; Annual Meeting, Providence, R. I., Sept. 14-17, 1926.

Trade Catalogs

The Kuhlman Electric Co., Bay City, Mich., manufacturer of Kuhlman power, distribution and street lighting transformers, announces the appointment of H. F. Darby, Jr., 1700 Walnut St., Philadelphia, Pa., as direct factory representative in the Philadelphia district.

Dual Clearance Una-flow Engines—The Farnsworth Co., engineers, manufacturers and contractors, Harrisburg, Pa., describes in bulletin No. 20 the dual clearance Una-flow Harrisburg heavy duty type engine, built from Stumpf patents.

Spiral Riveted Pipe—The American Spiral Pipe Works, Chicago, Ill. Illustrated folder on the Taylor spiral riveted pipe now made of copper bearing steel. Announcement is made of the flange catalog, No. 24, and the spiral riveted pipe book, No. 22.

Portable Electric Pipe Machines—Hall-Will, Inc., Erie, Pa., manufacturer of pipe, bolt and nipple threading machines, will mail on request to anyone interested the twelve-page catalog and price list illustrating and describing in detail the new Red-E-Hall portable electric pipe machine manufactured by the company.

Rod Bushings and Brasses—J. Faessler Manufacturing Co., Moberly, Mo., has issued Bulletin H on the Faessler-Maupin process for applying side and main-rod bushings and brasses made by the company. It is claimed that the Faessler-Maupin process produces an initial compression of the metal that cannot be increased by wear, which results in a perfect bearing surface of accurate bore and unique self-lubricating qualities.

Fuel Prices

COAL

The following table shows the trend of the spot steam market in various coals, f.o.b. mines; mine run except Pittsburgh gas slack:

Bituminous Net Tons	Market Quoting	Aug. 23 1926
Pool 1	New York	$2.40@$2.75
Smokeless	Boston	2.59
Clearfield	Boston	1.70@ 2.00
Somerset	Boston	1.85@ 2.10
Kanawha	Columbus	1.50@ 1.75
Hocking	Columbus	1.40@ 1.75
Pittsburgh	Pittsburgh	1.60@ 1.90
Pittsburgh gas slack	Pittsburgh	1.20@ 1.30
Franklin, Ill.	Chicago	2.35@ 2.50
Central, Ill.	Chicago	2.15@ 2.25
Ind. 4th Vein	Chicago	2.15@ 2.35
West Ky.	Louisville	1.10@ 1.35
S. E. Ky.	Louisville	1.50@ 1.85
Big Seam	Birmingham	1.75@ 2.00
Anthracite Gross Tons		
Buckwheat No. 1.	New York	1.75@ 3.50
Buckwheat No. 1.	Philadelphia	2.25@ 2.75
Birdseye	New York	2.00

FUEL OIL

New York—Aug. 26, light oil, tank-car lots; 28@34 deg. Baumé, 6c. per gal.; 36@40 deg., 6¾c. per gal. f.o.b. Bayonne, N. J.

St. Louis—Aug. 18, tank-car lots, f.o.b. St. Louis; 24@26 deg., $2.10 per bbl.; 26@28 deg., $2.15 per bbl.; 28@30 deg., $2.20 per bbl.; 30@32 deg., $2.25 per bbl.; 32@36 deg., gas oil, 6.37½c. per gal.; 38@40 deg., 7¼c. per gal.

Pittsburgh—Aug. 16, f.o.b. local refinery; 30@34 deg., fuel oil, 5.87½c. per gal.; 36@40 deg., fuel oil 6½c. per gal.

Dallas—Aug. 14, f.o.b. local refinery 26@30 deg., $1.75 per bbl.

Philadelphia—Aug. 19, 27@30 deg., $2.52@$2.58 per bbl.; 13@19 deg., $1.59 @$1.65 per bbl.

Cincinnati—Aug. 16, tank-car lots f.o.b. local refinery, 24@26 deg., Baumé, 6½c. per gal.; 26@30 deg., 6⅝c. per gal.; 30@32 deg., 7c. per gal.

Chicago—Aug. 23, tank-car lots f.o.b. Oklahoma, freight to Chicago, 92c. per bbl.; 24@26 deg., $1.40@$1.42½ per bbl.; 26@30 deg., $1.45@$1.50; 30@32 deg., $1.55.

Boston—Aug. 23, tank-car lots f.o.b. 12@14 deg. Baumé 4.48c. per gal.; 28@32 deg., 5.9c. per gal.

New Plant Construction

Calif., Brentwood — Brentwood Water District plans installing fire protection system and construction of deep well pumping plants. Estimated cost $20,000.

Calif., Fresno—Southern Pacific R.R. Co., 65 Market St., San Francisco, is having plans prepared for the construction of an ice plant, here. Estimated cost $150,000. G. W. Boschke is chief engineer.

Calif., Los Angeles—Warner Bros., c/o G. A. Lansburgh, Archt., Consolidated Bldg., are receiving bids for the construction of stores, office and theatre building at Wilcox and Hollywood Blvd. Estimated cost $2,000,000.

Calif., Miranda—South Fork Union High School Dist. is having plans prepared for the construction of high school including electrical power plant and water pumping station. Estimated cost $150,000. W. H. Weeks, 369 Pine St., San Francisco, is architect.

Calif., Nestor—Tia Juana River Irrigation District plans election to vote $220,000 bonds for one 1,100 g.p.m. one 700 g.p.m., 150 ft. direct connected centrifugal pumping plants; reservoir, pipe lines, etc. J. F. Covert, Box 533, National City, is engineer.

Calif., Tracy—Banta-Carbona Irrigation District plans an election to vote $340,000 bonds for installing 5 additional 60 sec. ft. pumps for stand-by system, concrete lining canals, etc. T. H. Means, 369 Pine St., San Francisco, is engineer.

Calif., Turlock—Turlock and Modesto Irrigation Districts plan 2 additional 7,500 kw. units for Don Pedro Power Plant.

Calif., Turlock—Turlock Irrigation District is having plans prepared for the construction of a power development, including power plant to develop 27,000 hp. with 3 turbine water wheels, 9,000 hp. each, flume, etc. Estimated cost $5,000,000. R. V. Meikle is chief district engineer.

D. C., Washington — U. S. Veterans Bureau, Arlington Bldg., will receive bids until September 14, for a new boiler at U. S. Veterans Hospital 32, at Mount Alto, here. L. H. Tripp is chief of construction division.

Ill., Chicago—Van Buren Building Corp., c/o I. M. Mills, 111 West Monroe St., is having plans prepared for the construction of 16 story, office and stores building, at Wells and Van Buren Sts. Estimated cost $1,500,000. D. S. Klafter, 64 North Randolph St., is architect.

Ill., Chicago—C. Wishnack, 112 South Wells St., is having plans prepared for the construction of 18 story apartment, at 5861-63 Sheridan Rd. Estimated cost $1,500,000. McNally & Quinn, 8 South La Salle St., are engineers.

Ill., South Beloit (mail Beloit)—Chicago, Milwaukee & St. Paul Ry., Union Station, Chicago, is receiving bids for the construction of timber and rein.-con. coal pit, with conveyor system, 125 ton capacity. Estimated cost $30,000. C. F. Loweth, Union Station, Chicago, is engineer.

Ind., Ft. Wayne—Fox Realty Co. is receiving bids for the construction of 8 story hotel and theatre, at Harrison and Jefferson Sts. Estimated cost $1,250,000. A. M. Strauss, Ft. Wayne, is architect.

Ind., Garrett—Common Council will receive bids until Sept. 7, for two 300 hp. water tube boilers, 2 semi mechanical stokers, 2 boiler feed pumps, etc. for improving municipal electric light and waterworks plant. A. U. Bower, clk.

Ind., Indianapolis—School Board, Indianapolis, awarded contract for the construction of a power house at P. S. 47, to A. H. Unversaw, Indianapolis. Estimated cost $35,000.

Ia., Silbey—City plans installing 400 hp. electric power unit.

Ia., Sioux City—A. Sanford, c/o Iowa Loan & Investment Co., 1200 State Lake Bldg., will soon receive bids for the construction of 8 story, theatre and office building at 6th and Pierce Sts. Estimated cost $1,800,000. C. W. and G. L. Rapp, 190 North State St., Chicago, Ill., are architects.

Kan., Oswatomie—Plans an election to vote $55,000 bonds for the construction of waterworks and power plant improvements. Previous issue defeated. Burns & McDonnell Engr. Co., 402 Mutual Bldg., is engineer.

La., New Orleans—Sewage and Water Board, 526 Carondelet St., will receive bids until October 15, for transformers for power house No. 2. A. G. Moffat is secretary.

Md., Baltimore—Mayor and City Council, City Hall, awarded contract for the construction of college building and power house, at 33rd St. and Alameda Blvd., to H. Miller, Inc., Eutaw and Franklin Sts. $1,980,000.

Mass., Brockton—Edison Electric Illuminating Co., 32 Boylston St., Boston, awarded contract for the construction of 1 story switchhouse, on Dupont Circle to Stone & Webster, 147 Milk St., Boston. Estimated cost $40,000.

Mass., Fitchburg — Burbank Hospital, Nicholas St., is receiving bids for the construction of 3 story ward building, with 1 story power house and laundry. To exceed $150,000. J. D. Leland & Co., 41 Mt. Vernon St., Boston, are architects.

Mass., Holyoke—Germania Mills are having plans prepared for rebuilding 1 story boiler house, including 2 new 600 hp. boilers, stokers, etc., on Main St. Estimated cost $60,000. McClintock & Craig, 458 Bridge St., Springfield, are engineers.

Mich., Detroit—Detroit Community Fund, 316 East Jefferson Ave., is having plans prepared for the construction of 4 story office building, with steam heating system, on Warren Ave. Estimated cost $800,000. Donaldson & Meier, Penobscot Bldg., are architects.

Mich., Detroit—Detroit Hockey Club, Inc., J. L. Woods, 615 West Hancock Ave., is having plans prepared for the construction of 1 story arena building, including steam heating system, on McGraw Ave. Estimated cost $1,200,000. C. H. Crane, Griswold St., is architect.

Mo., Kansas City—Henrici Lowry Eng. Co., 402 Security Bldg., Engrs. will soon receive bids for the construction of 3 story power plant, for Durks & Son Lumber Co., 19th and Woodland Ave.

Neb., Fremont—City is having plans prepared for the construction of a 30 x 100 x 50 ft. power plant addition. Estimated cost $100,000. Owners are in market for 600 hp. boiler. Burns & McDonnell Engr. Co., 402 Interstate Bldg., Kansas City, Mo., is engineer.

N. H., Tilton—Tilton School is receiving bids for the construction of 1 story power house. Estimated cost $40,000. Hutchins & French, 11 Beacon St., Boston, Mass., are engineers.

N. Y., New York—S. Augenblick, c/o M. Zipkes, 25 West 43rd St., Archt., is having plans prepared for the construction of office building on Lafayette St. Estimated cost $500,000.

N. J., Asbury Park—New Jersey Light & Power Co., Highstown, awarded contract for the construction of 11 story, office building here, to D. P. Robinson & Co., 61 Broadway, New York City, $750,000.

N. J., Hawthorne—Boro of Hawthorne is having plans prepared for the construction of a pumping station, for increasing water supply. Estimated cost $30,000. H. J. Harder, Market St., Paterson, is engineer.

N. J., Newark—Department of Parks and Public Safety, City Hall, will soon receive bids for the construction of power house, in connection with extensions to city hall. Estimated cost $200,000. J. S. Pigott, 402 Broad St., is architect.

N. Y., New York—Twenty-five East 77th Street Co., Inc., c/o Schwartz & Gross, 347 5th Ave., Archts. is having plans prepared for the construction of 15 story apartment, at Madison Ave. and 77th St. Estimated cost $1,000,000.

N. Y., New York—Fur Zone Realty Co., c/o H. I. Oser, Archt., 1440 Bway., awarded contract for the construction of 17 story office building, 252 West 30th St., to Magoba Constr. Co., Inc., 103 Park Ave. Estimated cost $1,500,000.

O., North Canton (br. Canton)—City is having plans prepared for the construction of waterworks system, including pumps, wells, pumphouse, etc. $50,000 to $60,000.

Okla., Rush Springs—City, L. Leverett, Mayor, is in the market for 100 g.p.m. motor driven triplex pumping unit for waterworks.

Pa., New Castle—Castleton Hotel is receiving bids for the construction of 6 story hotel, here. Estimated cost $750,000. W. G. Eckles & Son, New Castle, are architects.

Pa., Philadelphia — C. Siegel, Garden Court Apartments, is having plans prepared for the construction of 10 story and basement, apartment, at Pine and 47th St. R. Benckert, 13th and Chestnut Sts., is architect.

Pa., Phila. — Arcadian Court Inc., c/o G. B. Johnson, 237 North 16th St., awarded contract for the construction of 10 story apartment at 68th and Chestnut Sts., to Van Asdler Constr Co., Stock Exchange Bldg. Estimated cost $2,000,000.

Pa., Phila. — City, c/o Dept. Public Welfare, will receive bids until September 3, for the construction of a group of buildings, including a boiler house, at House of Correction, Holmesburg, here. P. H. Johnson, Widner Bldg., is engineer.

Pa., Wayne—United Gas & Improvement Co., Broad and Arch Sts., Philadelphia, awarded contract for the construction of 1 story boiler house here, to F. G. Stewart Co., 1520 Locust St., Philadelphia.

R. I., East Greenwich—A. H. Esten, Division St., awarded contract for the construction of dye and boiler house, on Division St., to Lock & Thomas, Inc., Olympia Bldg., New Bedford, Mass. Estimated cost $40,000.

Tex., Harlingen—City will receive bids after October 1, for 1,500 g.p.m., 200 ft. head pump; 750 hp. oil engine, direct connected to generator, mains, etc. Estimated cost $60,000. E. C. Bennett is superintendent of Public Utilities.

Tex., Houston—Houston Terminal Warehouse & Cold Storage Co., R. H. Baker, Pres., is having plans prepared for the construction of a 6 story cold storage plant, with 8 slow speed electric elevators, automatic ventilators, electrical equipment, etc. on Upper Ship Canal, here. Estimated cost $2,800,000. Engineering Service Corp., Houston, in charge of design and construction and Moores & Dunford, 100 East 42nd St., New York City are consulting engineers.

Tex., Waco—Waco Development Co., c/o A. Sanger, Pres., is having plans prepared for the construction of 18 story store and office building with 2 passenger and 1 freight elevators, at Austin and 4th Sts. Estimated cost $700,000. M. W. Scott & Co., 412½ Franklin St., are architects.

Wash., Seattle—Owner c/o E. Guenther, Railway Exchange Bldg., Archt., is having plans prepared for the construction of 10 story, apartment at Madison and Terry Aves. Estimated cost $750,000.

Wis., Berlin—Wisconsin Power & Light Co., 16 North Carroll St., Madison, awarded contract for the construction of power house and plant here, to A. J. Kramp Constr. Co., Berlin. Estimated cost $50,000.

Wis., Chippewa Falls — Clover Leaf Creamery, 417 Bridge St., is in the market for refrigerating machinery for proposed creamery, here.

Wis., Janesville — Wisconsin Power & Light Co., Gay Bldg., Madison, awarded contract for the construction of 1 story central hydro electric plant, to J. P. Cullen & Son Co., 109 South Main St. Estimated cost $100,000.

Wis., Shawano—Wisconsin Power & Light Co., 16 North Carroll St., Madison, is having plans prepared for the construction of rein.-con. dam on Wolff River. Estimated cost $50,000. Mead & Seastone, Journal Bldg., Madison, are engineers.

N. Z., Wellington—Public Works Department will receive bids until September 28, for 2 control boards complete for Arapuni Electric Power Scheme.